POWER and THE ENGINEER

EVERYTHING ABOUT THE GENERATION AND TRANSMISSION OF POWER

Vol. 32 NEW YORK, APRIL 19, 1910 No. 16

FREQUENTLY, it has been charged, POWER is too technical.

For those who believe this charge we are sincerely sorry.

Suppose we never printed anything except words of one syllable. How fast would your vocabulary increase? Suppose we never printed anything except what you already knew. How fast would your store of learning increase? Suppose we printed nothing except what you could easily understand—say, things which were obvious to you on account of such information as you already possessed. To you, wherein would the value of POWER lie?

On the other hand, suppose we put before you some facts, part of which you knew to be true, and said that such and so being true, it follows that certain results must occur. Now, as only part of the facts were known by you to be true, suppose that it was not clear to you that the conclusions which we drew, were correct. What would you do?

One man might throw up his hands in holy horror, exclaiming, POWER is too technical for me! In other words, admit that he was whipped before he attempted even to "take a fall" out of the "mysteries" which the article contained.

Another chap may reason as follows: POWER says that such and so happens; I don't exactly see how it can; but then, POWER can't afford to give me incorrect information, I'll take a chance and assume the facts to be as stated.

But a third fellow says: Now, let's see. I know that part of the statements made here are true because that's the way I have "doped" them out, but as to these other assertions and the conclusions which are drawn, I am rather at a loss to understand how they can be proved to be so.

Forthwith, he gets after further information. He goes to the reference books which he owns. If they cannot help him, he questions friends. If they are unable to show him, he becomes all the more anxious to find the answer, for, if he can get it, he will then know just so much more than they.

So he writes to POWER and puts it up to us.

This last man is a winner. He wants what he wants. Such men usually get there.

He is quite right in putting it up to us too. Our business is to disseminate information. If we don't get it to you in a digestible form, we want you to "jack us up" about it. If we don't give you all the information you need or may want, we are the ones to whom you should make the complaint.

Don't be like the first or second of the men we just spoke about.

If a subject interests you and you read an article in POWER on it which isn't quite as clear as you might want it to be, put it up to us. We'll be glad to hear from you.

Norfolk's Traction Power Plant

A Handsome Turbine Plant Kept in the Pink of Condition. Coal Handling Facilities by Water or Rail a Feature of Special Interest

BY WARREN O. ROGERS

Not all of the best-designed, well managed or richly finished power plants are found in the cities of the Northern States. Although having visited hundreds of steam plants, the writer has yet to see one that will appeal more forcibly to the artistic as well as to good engineering than will the turbine room of the power plant of the above-named company, which is located at Brambleton, in the City of Norfolk, Va.

The building is constructed of hard-pressed red brick, is 158 feet 6 inches wide and 169 feet 6 inches long, on the boiler-room side; on the turbine-room side it is 148 feet long, the shorter end appearing as shown in Fig. 2. The interior walls are lined with white enamel brick to the hight shown in Fig. 1. The wainscoting above the switchboard gallery, and the turbine pit are lined with brick of the same finish. The flooring is made of small blocks of hexagonal red tile. The top railing and tops of the posts around the switchboard and turbine platform are constructed of polished brass, as are also the conduit pipes leading from the turbine generator to the floor, through which the cables are carried to the switchboard underground. Over the turbine are located skylights which make an exceedingly light turbine room. A track is constructed in the floor-ing for about one-third the length of the turbine room. The plant is kept exceptionally clean and the machinery and piping maintained in the best of condition.

FIG. 1. TURBINE ROOM SHOWING THREE 3000-KILOWATT AND ONE 1500-KILOWATT CURTIS STEAM TURBINES

TURBINE ROOM

A general view of the turbine room, taken from the switchboard end and side, is shown in Fig. 1. Here are located one 1500-kilowatt and three 3000-kilowatt Curtis turbine-driven generators, running at a speed of 900 and 720 revolutions per minute, respectively. The current from each machine is sent to the transformers at 11,000 volts. Each turbine unit is set on an Alberger surface condenser, and the 1500-kilowatt unit is piped to an Alberger 8 and 20 by 12-inch dry-vacuum pump, the circulating water being furnished by a 16-inch Alberger centrifugal pump, direct connected to a vertical steam engine.

The three 3000-kilowatt units are each connected to a 12 and 22 by 18-inch Alberger pump, the circulating water being supplied by a 20-inch centrifugal pump, each being direct connected to a vertical steam engine. The water from the hotwell is handled by a 6 and 8¼ by 12-inch

Alberger steam-driven pump, in the case of the small turbine, and by three 8 and 12 by 12-inch steam-driven vertical Alberger pumps for the other three units. The locations of these pumps and of the oil step-bearing pumps are shown in the plan view, Fig. 3, and the sectional view, Fig. 4. The four 9x8⅝x5⅛x10-inch Blake oil step-bearing pumps supply the step bearings with oil at a pressure of 900 pounds per square inch. On the same side of the turbine room are located four Dean 18 and 12 by 24-inch vertical type boiler-feed pumps. The auxiliaries exhaust their steam into a feed-water heater located between No. 1 and No. 2 turbines. The auxiliaries are all in sight of the engineers, the oil pumps, boiler-feed pumps and exciters being set under the gallery on the boiler-room side of the station.

Exciters

One exciter set consists of a 25-kilowatt, 125-volt continuous-current generator, direct connected to a 25-kilowatt noncondensing Curtis turbine of the horizontal type, which runs at a speed of 3600 revolutions per minute under a steam pressure of 175 pounds per square inch. There are also two 75-kilowatt direct-current exciters delivering current at 125 volts, each direct connected to a horizontal Curtis turbine of 75 kilowatts capacity, running at a speed of 2400 revolutions per minute.

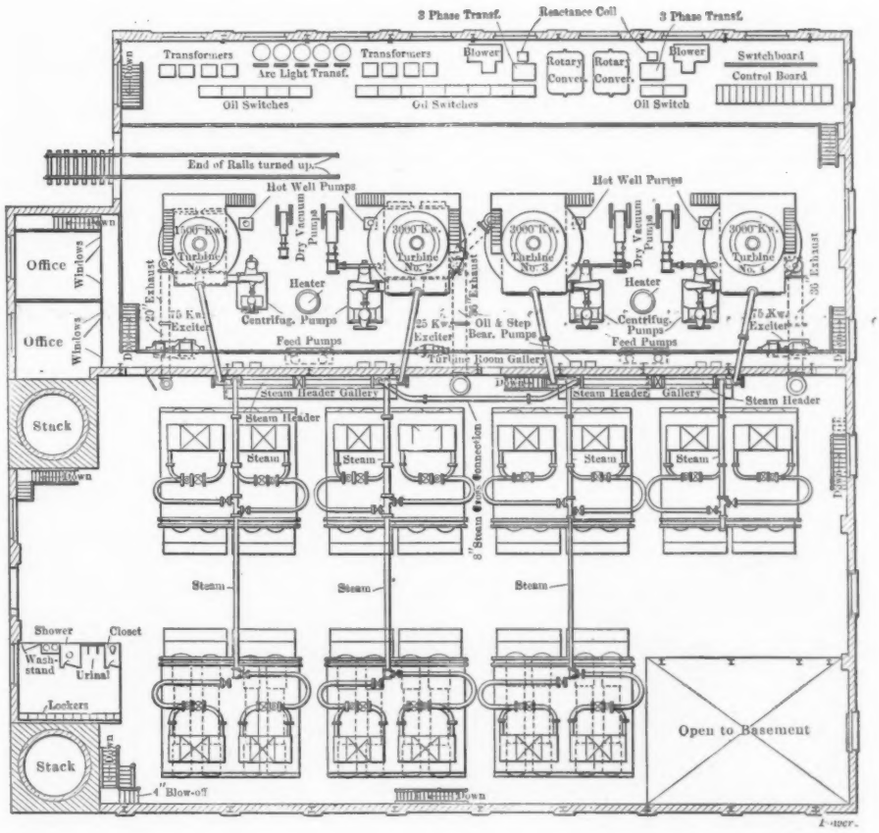


Fig. 3. Plan View of the Norfolk and Portsmouth Traction Company's Power Plant

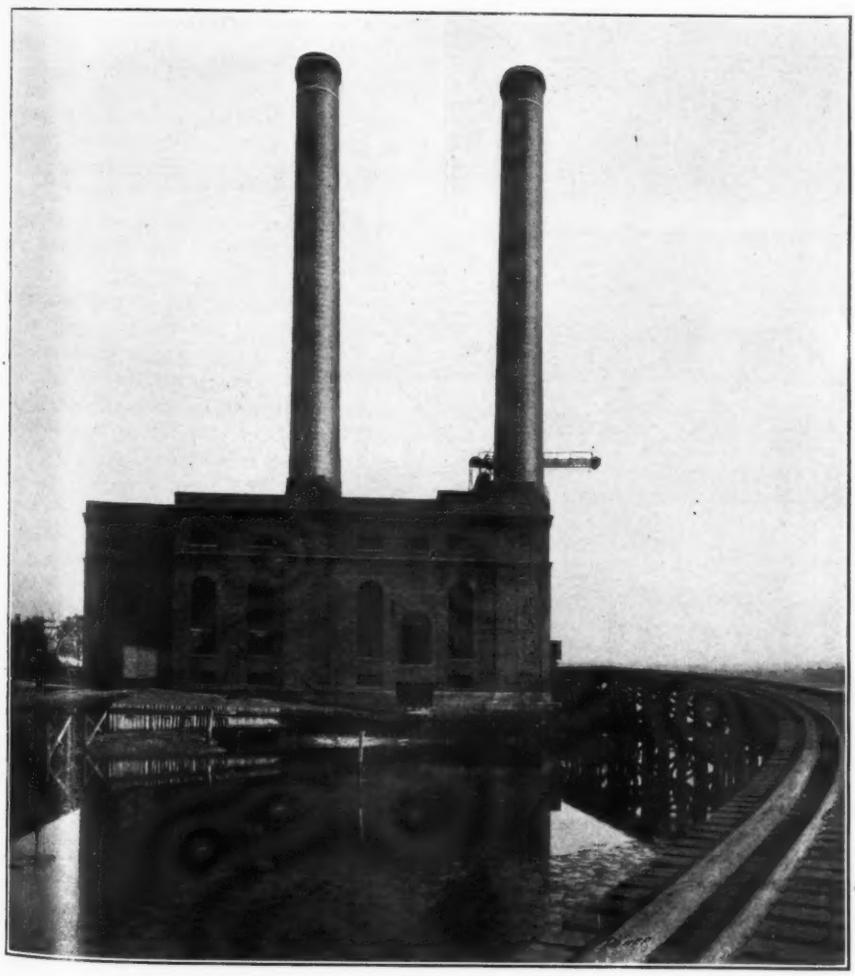


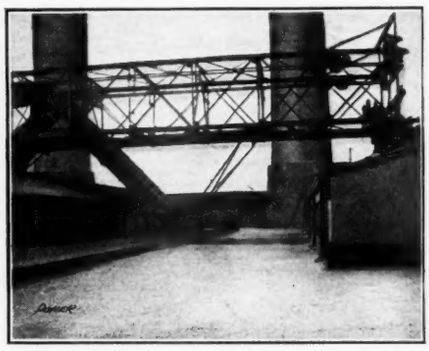
Fig. 2. End View of Power Plant Showing Source of Condensing Water and Intake

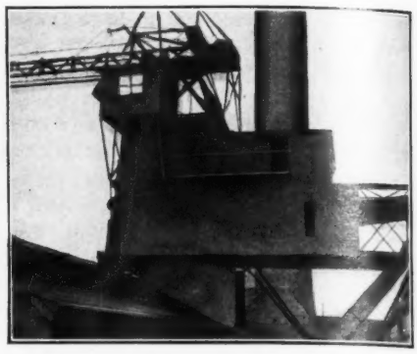
Boilers

The arrangement of the boilers is shown in Figs. 3, 4 and 5, the latter being a general view between the two rows of boilers. The entire installation consists of fourteen 512-horsepower Babcock & Wilcox boilers, which are fitted with superheaters of the Babcock & Wilcox type. The boilers are arranged as shown in Fig. 4. Each has 252 four-inch tubes, 18 feet long. They are 18 feet wide and 14 feet high, and have two 42-inch drums made of open-hearth steel of 56,000 pounds tensile strength; they are 23 feet 7 3/16 inches long. The boilers are designed to carry a steam pressure of 200 pounds per square inch, and contain 5000 square feet of heating surface. The smoke flue, running the entire length of both rows of boilers is arranged as shown in Fig. 4, and is connected to two separate brick stacks, both being 12 feet 6 inches in diameter and 200 feet high above the boiler-room floor. The boilers are equipped with Roney stokers, which are driven by a Westinghouse stoker engine of standard make.

Piping

A study of the plan and elevation drawings, Figs. 3 and 4, will give a good idea of the general arrangement of the piping. Fig. 3 shows the arrangement over the boilers and the connections to the headers which are placed on the boiler-room side of the partition wall. As the dimensions are given in Fig. 4, and the general arrangement from both Figs. 3 and 4 is clear, little remains to be said regarding the pipe-line design. It should be noted that the piping pertaining to the turbine units and auxiliaries is placed in the basement under the turbine-room floor and that the blowoff and feed pipes are placed in the basement under the boilers. From Fig. 4 it will be seen that the condensing water is taken from the intake tunnel, at a lift of 22 inches, the inlet to the tunnel and source of condensing-water supply being shown in Fig. 2. Each turbine is connected to an atmospheric exhaust pipe, the 1500-kilowatt unit having a 20-inch pipe, and each of the 3000-kilowatt units a 36-inch exhaust pipe, all being equipped with suitable exhaust heads.





Figs. 6 and 7. Two Views of the Coal-handling Apparatus

Fig. 5. The Firing Alley in the Boiler Room

Coal Delivery

The method of handling the coal supply at this plant is somewhat out of the ordinary. It is possible to deliver it by water or by rail, the latter being the more convenient; but if rail shipments should be delayed, cargoes of coal can be unloaded at the wharf. Generally, the coal is delivered in cars which are run in over the coal bin on the track shown in Fig. 2, and the cars are unloaded through the bottom hoppers onto the coal wharf.

Running on rails laid on the two walls

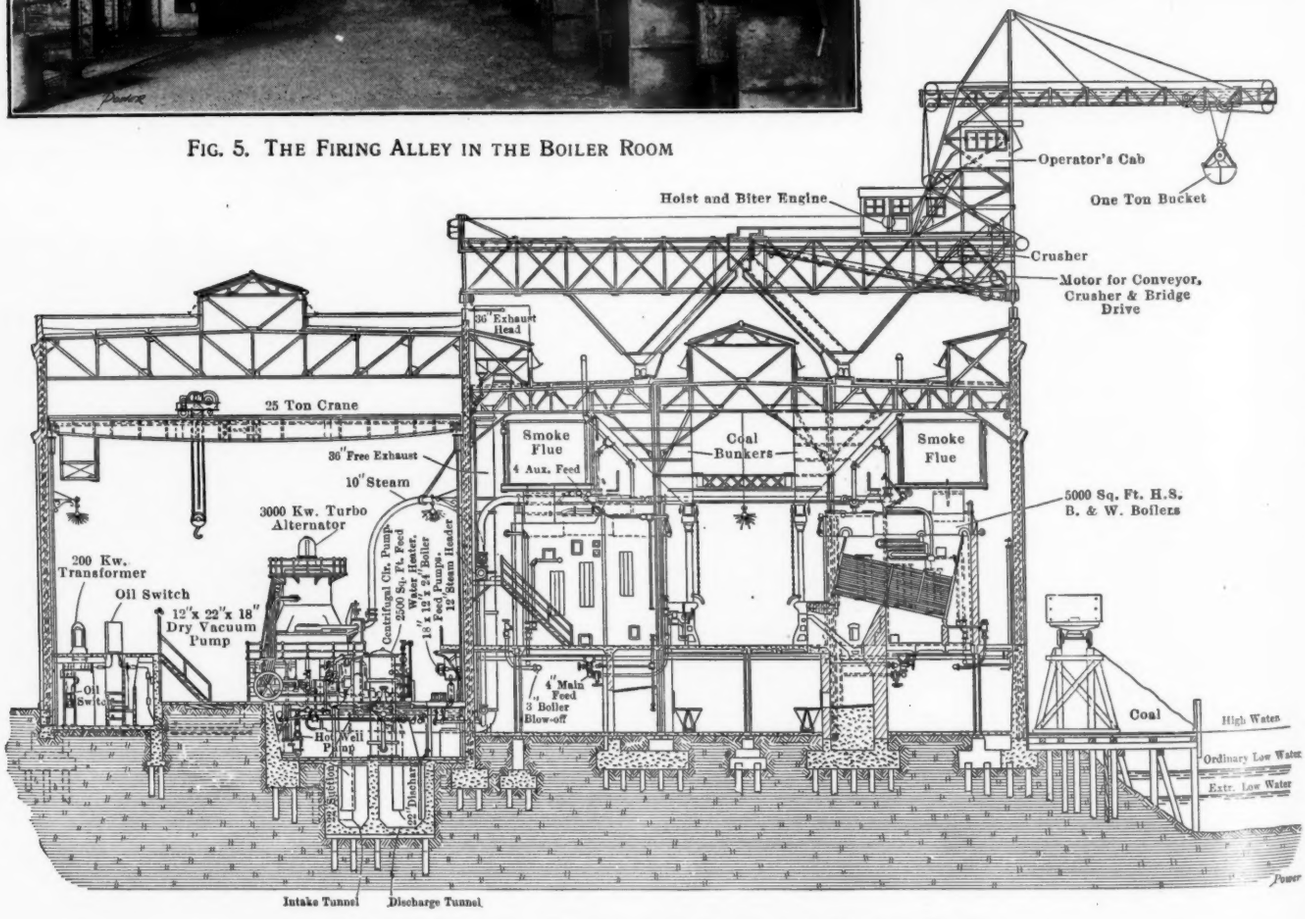


Fig. 4. Sectional Elevation of Boiler and Turbine Rooms

forming the sides of the boiler room, which extend up about 12 feet above the roof, is a bridge drive on which are arranged the necessary operating motors, the coal-crushing rolls, the one-ton bucket and the conveyer belt. Through the roof of the boiler room is an opening to the coal bins, which are placed above the boilers. These bin walls are straight on the inner side, but slant at the rear and ends, as shown in Fig. 4. From the bottom, chutes deliver the coal to the hoppers of the stokers as required, the amount fed being regulated by chain-operated valves in the chutes.

The operation of the coal-handling system is simple. When coal is wanted, the operator in the cab of the bridge lowers the one-ton bucket which picks up its load of fuel from the coal wharf, and is

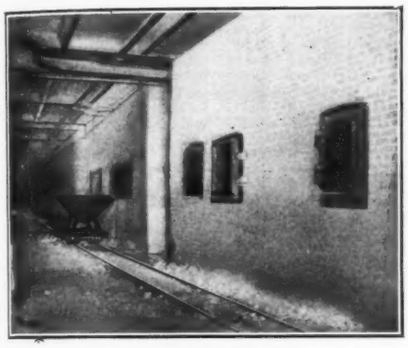


FIG. 8. ASH DELIVERY

hoisted up and conveyed up over the crushing rollers, on which it is dumped and, after passing through the rollers, is delivered to a belt conveyer and carried to a point midway between the two walls, or to practically the center of the bridge. Here the fuel is delivered to either one of the two chutes which deliver the coal to the bunkers. At the bottom of each chute is fitted a gate which is operated by means of a lever running to the operating cab. Covers are provided to close the openings to the coal bunkers if desired. As soon as one section has been filled the bridge is moved on to the next section requiring replenishment. Figs. 6 and 7 shows a near view of the trolley and coal chutes. The hoist is operated by a 100-horsepower 600-volt motor; the crusher and trolley motors are each of 25 horsepower capacity.

The ashes are received in an ashpit under the boilers and shoveled out into an ash car running on a narrow-gage track, extending the entire length of the boiler-room basement, as shown in Fig. 8. This track extends around the end of the boiler house to a point under the railroad trestle, where the ashes are dumped into a pit and afterward elevated by means of a chain-bucket conveyer to a coal car for shipment.

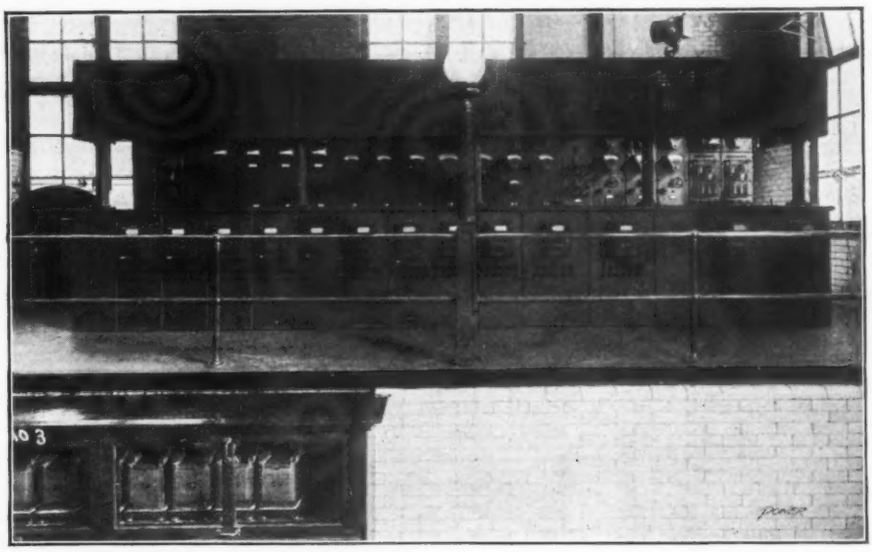


FIG. 9. SWITCHBOARD GALLERY

SWITCHBOARD

The switchboard is shown in Fig. 9. It is constructed of slate and is located on the extreme end of the gallery, as shown in Fig. 3. It contains the necessary instruments for recording the outputs of generators, also the switches, etc., for controlling the excitation current. The board controlling the oil switches is located on the same gallery, and the switches are below the gallery, as shown in Figs. 3 and 4. On the switchboard platform are located the transformers and oil switches, also two General Electric converters which are 60-cycle on the alternating-current side, with an output of 870 amperes at 575 volts on the direct-current side; both units run at 600 revolutions per minute.

Each turbine has a separate signal system, a signal board for each being placed and arranged as shown in Fig. 10. By pressing the two-way switch on either panel, say for No. 1 generator, if it is desired to stop, the switch opposite the word "stop" is pushed. This causes a bell to ring at the turbine and one at the switchboard, besides lighting a lamp under the word "stop" on both the switchboard and turbine-signal box. The other signals are given in the same way. In this particular plant, the switchboard operator has an unrestricted view of the entire turbine room from his post of duty.

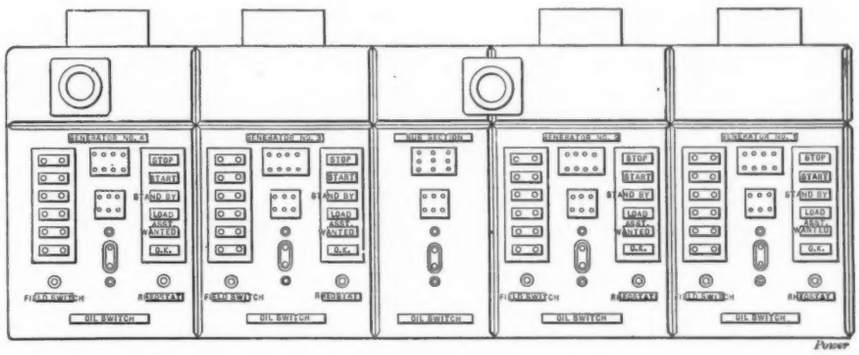


FIG. 10. DIAGRAM OF SIGNAL BOARD

Pat Callahan was the chief engineer of a large factory in which the firm had just installed a fine new Corliss engine. Pat was strictly onto his job, and when the erecting engineer had started up the engine and left, he had left the bearings all adjusted loose with the understanding that Pat would take them up gradually as they wore down to a smooth bearing. The result was that it pounded somewhat.

A few days later an oil salesman called on Pat, and after handing him one of the usual "Rubberino" brand of cigars, he remarked: "Pat, why the devil don't you key up that engine?"

Pat eyed the cigar suspiciously, bit off the end of it, and as he reached for a match, remarked: "Well, considerin' thot it's a new ingin, oi'd a domd soight ruther hear it than 't schmell it."

Feed Water Heaters of the Closed Type

Saving Due to Heating the Feed Water by Exhaust Steam. Condensation in Heater. Effect of Scale and Oil. Types of Closed Heater

BY WILLIAM F. FISCHER

There are two main sources of waste in all steam plants, the first being the heat escaping with the gases of combustion, the second, the heat in the exhaust steam escaping from the engines.

It has been stated that under the most favorable conditions a noncondensing plant can never realize more than 7 per cent. of the heat value of the fuel as power. Six per cent. represents good average practice in noncondensing plants and 10 per cent. in condensing plants; and as ordinarily burned under steam boilers in manufacturing plants, probably less than 5 per cent. is realized in effective work at the engine shaft. Certain types of furnaces, if properly handled, develop higher efficiencies than others, as they more completely reduce the fuel to combustible gases and burn the gases under more favorable conditions, thus securing complete combustion and greater absorption of the heat by the water in the boiler.

With boilers and furnaces of the most efficient type a large percentage of the heat of the fuel passes up the chimney with the products of combustion, and much of the heat of the fuel is utilized in changing water into steam. For example: It requires 966 B.t.u. to change one pound of water into steam at atmospheric pressure, while the temperature of this steam as shown by the thermometer is still 212 degrees Fahrenheit, the same temperature as the water.

This hidden heat, which the thermometer does not register, is called latent heat or the latent heat of vaporization.

Owing to the low thermal efficiency of the steam engine probably not more than 20 per cent. of the heat in the steam can be converted into mechanical power. Consequently most of the heat is necessarily rejected by the exhaust without doing work.

An example of this loss is shown in the low-pressure steam turbines, designed to operate in connection with reciprocating engines using the highly expanded steam from the engines before it flows into the condenser under a high vacuum.

Fuel is the greatest single factor in the cost involved in producing power, and represents from 50 per cent. in small stations to some 60 per cent. or more in the larger plants. Therefore, as all unused or waste-heat units cost just as much, in proportion to the price paid for fuel, as the useful units, the waste-heat units should, in the interest of economy, be again returned to the boiler with the feed water, and otherwise utilized to the best possible advantage.

t = Temperature of feed water in degrees Fahrenheit before heating.

H = Total number of B.t.u. above 32

TABLE 1. PERCENTAGE OF SAVING FOR EACH DEGREE OF INCREASE IN TEMPERATURE OF FEED WATER HEATED.

Initial Feed Temp.	Pressure of Steam in Boiler, Lb. per Sq.In. above Atmosphere.										Initial Feed Temp.
	20	40	60	80	100	120	140	160	180	200	
32°	0.0872	0.0861	0.0855	0.0851	0.0847	0.0844	0.0841	0.0839	0.0837	0.0835	0.0833	32°
40°	0.0878	0.0867	0.0861	0.0856	0.0853	0.0850	0.0847	0.0845	0.0843	0.0841	0.0839	40°
50°	0.0886	0.0875	0.0868	0.0864	0.0860	0.0857	0.0854	0.0852	0.0850	0.0848	0.0846	50°
60°	0.0894	0.0883	0.0876	0.0872	0.0867	0.0864	0.0862	0.0859	0.0856	0.0855	0.0853	60°
70°	0.0902	0.0890	0.0884	0.0879	0.0875	0.0872	0.0869	0.0867	0.0864	0.0862	0.0860	70°
80°	0.0910	0.0898	0.0891	0.0887	0.0883	0.0879	0.0877	0.0874	0.0872	0.0870	0.0868	80°
90°	0.0919	0.0907	0.0900	0.0895	0.0888	0.0887	0.0884	0.0883	0.0879	0.0877	0.0875	90°
100°	0.0927	0.0915	0.0908	0.0903	0.0899	0.0895	0.0892	0.0890	0.0887	0.0885	0.0883	100°
110°	0.0936	0.0923	0.0916	0.0911	0.0907	0.0903	0.0900	0.0898	0.0895	0.0893	0.0891	110°
120°	0.0945	0.0932	0.0925	0.0919	0.0915	0.0911	0.0908	0.0906	0.0903	0.0901	0.0899	120°
130°	0.0954	0.0941	0.0934	0.0928	0.0924	0.0920	0.0917	0.0914	0.0912	0.0909	0.0907	130°
140°	0.0963	0.0950	0.0943	0.0937	0.0932	0.0929	0.0925	0.0923	0.0920	0.0918	0.0916	140°
150°	0.0973	0.0959	0.0951	0.0946	0.0941	0.0937	0.0934	0.0931	0.0929	0.0926	0.0924	150°
160°	0.0982	0.0968	0.0961	0.0955	0.0950	0.0946	0.0943	0.0940	0.0937	0.0935	0.0933	160°
170°	0.0992	0.0978	0.0970	0.0964	0.0959	0.0955	0.0952	0.0949	0.0946	0.0944	0.0941	170°
180°	0.1002	0.0988	0.0981	0.0973	0.0969	0.0965	0.0961	0.0958	0.0955	0.0953	0.0951	180°
190°	0.1012	0.0998	0.0989	0.0983	0.0978	0.0974	0.0971	0.0968	0.0964	0.0962	0.0960	190°
200°	0.1022	0.1008	0.0999	0.0993	0.0988	0.0984	0.0980	0.0977	0.0974	0.0972	0.0969	200°
210°	0.1033	0.1018	0.1009	0.1003	0.0998	0.0994	0.0990	0.0987	0.0984	0.0981	0.0979	210°
220°		0.1029	0.1019	0.1013	0.1008	0.1004	0.1000	0.0997	0.0994	0.0991	0.0989	220°
230°		0.1039	0.1031	0.1024	0.1018	0.1012	0.1010	0.1007	0.1003	0.1001	0.0999	230°
240°		0.1050	0.1041	0.1034	0.1029	0.1024	0.1020	0.1017	0.1014	0.1011	0.1009	240°
250°		0.1062	0.1052	0.1045	0.1040	0.1035	0.1031	0.1027	0.1025	0.1022	0.1019	250°

Heating the Feed Water

Cold water should never be fed into a boiler, even if it becomes necessary to take live steam from the boilers to do the heating, as all steam boilers are more or less seriously affected by expansion and contraction strains. The contraction of certain tubes in water-tube boilers is likely to cause leakage, while in return-tubular boilers even more serious results may be expected, such as cracked plates, leaky seams, etc.

Water fed into a boiler cannot be converted into steam until heated from its original temperature to that corresponding to the steam pressure. If this water can be made to absorb heat which would otherwise go to waste through the engine exhaust, or in the flue gases, before it is fed into the boilers, it would be good economy to do so. Money must be expended for the necessary heating apparatus, piping, valves, fittings, building space and labor to operate the heaters, however, in order to effect this saving and the engineer should determine beforehand, as near as possible, what percentage of fuel will be saved by installing heat-saving apparatus.

If the steam pressure and feed-water temperatures before and after heating are known the approximate saving may be computed by the following formula:

$$\textit{Per cent. of fuel saved} = \frac{100\ (T - t)}{H + 32 -}$$

where

T = Temperature of feed water in degrees Fahrenheit after heating.

degrees Fahrenheit per pound of steam at the given boiler pressure.

The saving may be calculated directly from Table 1, which shows the percentage of saving for each degree of increase in temperature for various combinations of initial temperature of the feed water and working pressure in the boiler. This table is based on the above formula.

Rule: Multiply the factor in the table corresponding to any given initial temperature of feed water and boiler pressure by the total rise in feed-water temperature; the product will be the percentage of saving.

Example: Steam pressure equals 160 pounds. Initial temperature of feed water equals 50 degrees Fahrenheit. If the feed water is raised in the heater from 50 degrees Fahrenheit to 210 degrees Fahrenheit before entering the boiler, what is the percentage of saving?

Factor in table = 0.0850.

Rise in temperature = 210 — 50 = 160 degrees Fahrenheit.

$160 \times 0.0850 = 13.6$ *per cent.*

As an example, take the following: The temperature of steam at 165 pounds absolute pressure (150.3 pounds gage) is 365.7 degrees Fahrenheit. The total heat above 32 degrees Fahrenheit in a pound of water at this temperature equals 338.4 B.t.u.

The latent heat of vaporization, or the quantity of heat required to change one pound of water at 365.7 degrees Fahrenheit into steam at the same temperature equals 855.1 B.t.u.

The total heat in the steam (at 165 pounds absolute) is therefore

$$338.4 + 855.1 = 1193.5$$

B.t.u. per pound.

If the feed water enters the boiler at a temperature of 50 degrees Fahrenheit, each pound of the water must absorb

$$338.4 - (50 - 32) = 320.4 \text{ B.t.u.}$$

before it reaches the boiling point.

In order to entirely vaporize this water at 365.7 degrees Fahrenheit into steam at the same temperature, we must add 855.1 B.t.u. to every pound weight, in order to effect this change.

The total heat required to convert the feed water at 50 degrees Fahrenheit into steam at 165 pounds absolute pressure is, therefore,

$$320.4 + 855.1 = 1175.5$$

B.t.u. per pound.

Saving Due to Heating the Feed Water by Exhaust Steam

If, in the above case, the feed water were heated to about 210 degrees Fahrenheit, in an exhaust-steam feed-water heater before entering the boilers, the saving would be approximately 13.6 per cent., computed as follows:

$$\textit{Per cent. of fuel saved} = \frac{100\ (T - t)}{H + 32 - t} =$$

$$\frac{100 \times (210^\circ - 50^\circ)}{1193.5 + 32^\circ - 50^\circ} = \frac{16{,}000}{1175.5} =$$

13.6 *per cent.*

If, after leaving the heater the feed water were heated from 210 degrees Fahrenheit to say 300 degrees Fahrenheit in an economizer, the saving due to the economizer would be

$$\frac{100 \times (300^\circ - 210^\circ)}{1193.5 + 32^\circ - 210^\circ} = \frac{9000}{1015.5} =$$

8.86 *per cent.*

The combined saving, due both to the exhaust heater and the economizer, would be, according to the above,

$$13.6 + 8.86 = 22.46 \textit{ per cent.}$$

Since the heat supplied to the feed water in this manner is not taken from the fuel nor transmitted through the boiler plates, tubes, etc., the actual saving will be somewhat greater than the foregoing, as the fire is not forced so hard, resulting in better boiler and furnace efficiency.

The actual saving will average, according to good authority, about 1 per cent. for every eight or nine degrees aded to the temperature of the feed water in the heater, or from 1.2 to 1.4 per cent. for each 10 degrees rise.

Condensation in Exhaust Steam Heaters

As the exhaust steam is condensed in the feed-water heater it parts with its latent or hidden heat, which is absorbed by the cold feed water, tending to raise its temperature to that of the steam. The latent heat of steam at atmospheric pressure (14.7 pounds per square inch) is 966 B.t.u. Therefore, for every pound of exhaust steam condensed in the heater, 966 B.t.u. are given off or transferred from the steam to the water, and as each B.t.u. will raise the temperature of one pound of water one degree Fahrenheit, each pound of the steam thus condensed will raise 966 pounds of water one degree Fahrenheit, or

$$\frac{966}{211^\circ - 50^\circ} = 6$$

pounds of water from 50 degrees to 211 degrees Fahrenheit. The amount of condensation, therefore, amounts to about one-seventh of the weight of the feed water passing through the heater.

In a closed or pressure heater the condensation must be removed, as the steam and water are not in contact with each other. The water of condensation in a closed heater is usually trapped to the sewer, but it may be trapped to a receiving tank, combined with a pump, automatically controlled by the water level in the tank, and from thence returned to the boilers. If there is sufficient waste steam to heat the water to the desired temperature, however, this condensation in the heater is of no value, and would not pay for the cost of its recovery, as all oil and grease must be extracted from it before returning to the boilers.

Closed or pressure heaters may be grouped into two classes: Water-tube heaters, in which the water circulates through tubes, surrounded by exhaust steam, and steam-tube heaters, in which the exhaust steam circulates through the tubes, surrounded by the feed water.

Closed or pressure heaters are usually placed on the pressure side of the feed pumps, the pumps forcing the water through the heater and feed main into the boilers. All closed or pressure heaters are, or should be, equipped with a relief or safety valve to protect the heater from any overpressure.

Scale in Closed Heaters

Where feed water is free from acids, salts, sulphates and carbonates, so that no scale is formed under high temperatures, the closed heater will be found satisfactory.

This kind of water is not available in the majority of cases, however; therefore, all such water should be chemically or otherwise treated before entering the heater, as closed heaters are not well adapted to dirty or muddy feed water as the tubes soon become coated with mud or scale, impairing the transmission of heat and making frequent cleaning a necessity.

Although part of this scale and mud adheres to the heating surface of the heater the remainder may be swept into the boiler with the feed water, fastening itself to the boiler plates and tubes.

In heating the feed water to 210 or 212 degrees Fahrenheit the scale-forming substances that become insoluble at these temperatures are precipitated in the heater.

Therefore heaters of the water-tube type should be equipped with ample sediment chambers and may be thus used with water containing organic or earthly matter, but not scale-forming ingredients.

Carbonate of lime, for instance, is liable to combine with earthy matter and form an exceedingly hard scale, which soon coats the tubes to a considerable depth.

Unless closed heaters are kept clean they soon commence to run down in capacity and efficiency due to these scale deposits on one side of the heating surface, and the accumulation of oil and grease from the exhaust steam on the other.

A feed-water filter and settling tank, or chemical purifying system, is often used to purify the feed water, remove all sediment, or stop the largest part of it before using.

Table 2 shows the approximate percentage of lime precipitated from feed water when heated to various temperatures.

TABLE 2. PERCENTAGE OF LIME PRECIPITATED FROM WATER AT VARIOUS TEMPERATURES.

Temperature of Water in Degrees Fahrenheit.	Percentage of Lime Thrown Down.
217	50.0
219	52.3
221	56.8
227	60.5
232	64.5
236	69.0
240	73.3
245	77.4
250	81.7
255	86.0
261	90.3
266	94.0
271	97.7
290	100.0

Oil in Closed Heaters

In passing through the engines and auxiliary cylinders, the steam picks up small particles of lubricating oil, which are carried by the exhaust steam, in the form of emulsion or vapor, and deposited on the heating surfaces.

The tubes in time become thickly coated with this oil, thus reducing the efficiency of the heater to a considerable extent, besides in some cases causing pitting and corrosion. An oil separator should be placed in the exhaust main to intercept as much of this oil as possible before the exhaust steam reaches the heater.

As the exhaust steam and the feed water do not mingle in heaters of the closed type there is no danger of contaminating the feed water by impurities in the steam.

Coil and Bent-tube Heaters

Heaters of the coil type, having no sediment chambers, should not be used with waters that will precipitate sediment or scale-forming matter. The coil heater is very efficient as a heater, however, as the water which circulates through the coils is a long time in contact with the surfaces surrounded by and heated by the exhaust steam.

Straight-tube heaters having sediment chambers are more readily cleaned than those having curved or bent tubes, although the latter allow more freedom for expansion and contraction. It is extremely difficult to remove hard scale from the inside of curved or bent tubes.

Closed heaters are sometimes rated on the basis of one-third of a square foot of heating surface per horsepower. A 1500-horsepower heater would, therefore, contain 500 square feet of heating surface.

Vacuum or Primary Heaters

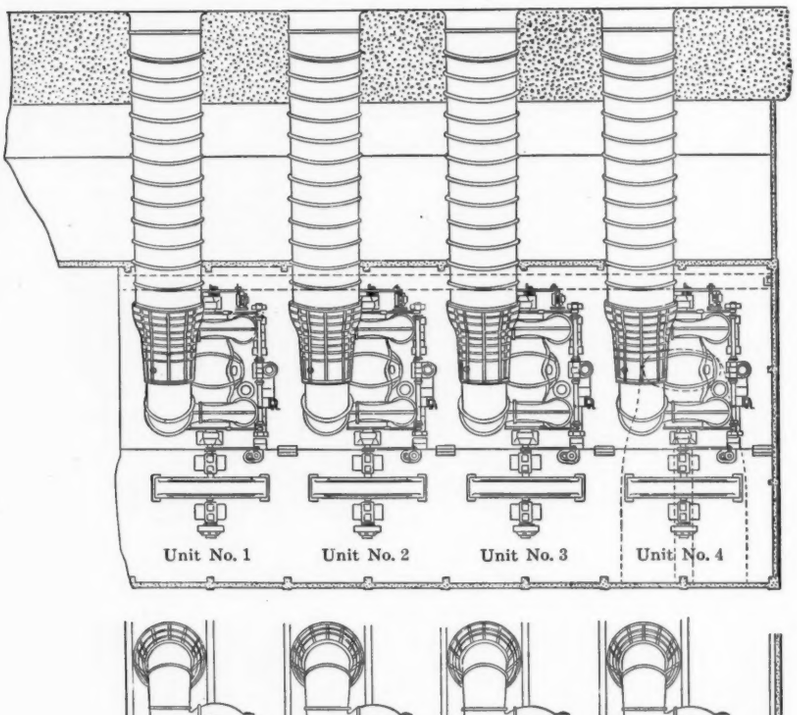
What is sometimes known as a vacuum or primary heater is merely a feed-water heater of the closed or pressure type, placed in the exhaust line between the engine and condenser, and extracting heat from the exhaust steam before the steam passes into the condenser. The feed water passing through this type of heater will reach about 108 to 110 degrees Fahrenheit, and can then be delivered to an atmospheric exhaust heater where its temperature is raised to about 210 degrees Fahrenheit, and from thence to an economizer, if so desired, increasing its temperature still higher before entering the boilers. As the vacuum heater acts as a condenser it should be absolutely tight, as a small air leak in the heater or exhaust pipe may seriously impair the vacuum in the condenser. This type of heater is not used to any great extent, as, ordinarily, there is sufficient exhaust steam at atmospheric pressure, from the steam-operated auxiliaries, to heat the feed water in an atmospheric heater.

Four 9000-Horsepower Water Turbines

Turbines Being Installed in the Little Falls Plant of the Washington Power Company Show Important Advance. Increased Efficiency and Volume

*BY H. BIRCHARD TAYLOR**

The four hydraulic turbines now being installed in the Little Falls plant of the Washington Water Power Company, on the Spokane river, Washington, are among the largest in the world. Each turbine is designed to have a capacity to deliver to the generator which it drives, 9000 mechanical horsepower when operating under a head of 66 feet and when running at a speed of 150 revolutions per minute. For these conditions of power and head, each turbine will require about 1500 cubic feet of water per second.



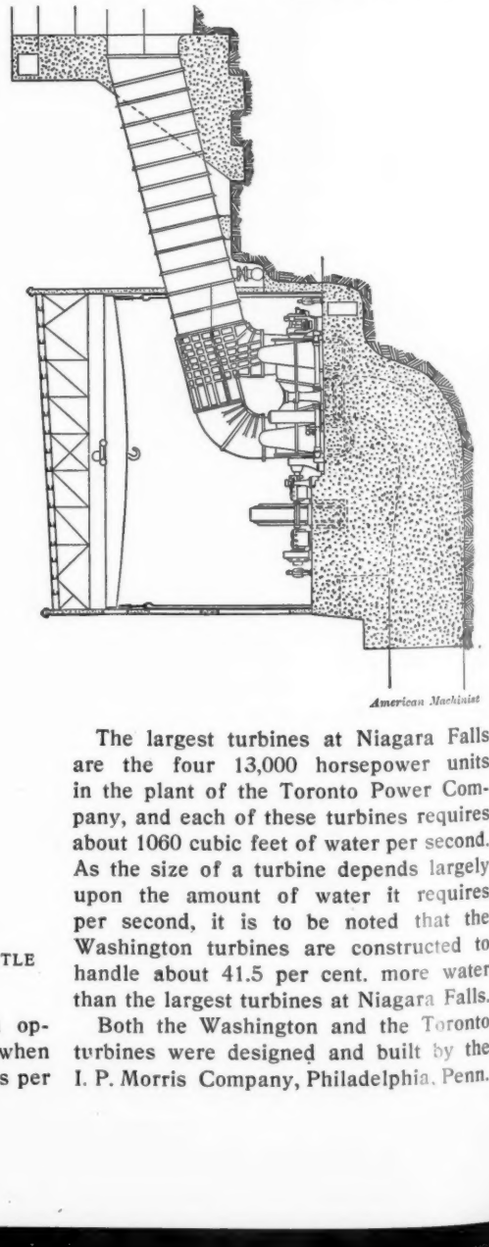


Fig. 1. Power House Arrangement of 9000-Horsepower Turbines at Little Falls, Washington

The largest turbines at Niagara Falls are the four 13,000 horsepower units in the plant of the Toronto Power Company, and each of these turbines requires about 1060 cubic feet of water per second. As the size of a turbine depends largely upon the amount of water it requires per second, it is to be noted that the Washington turbines are constructed to handle about 41.5 per cent. more water than the largest turbines at Niagara Falls.

Both the Washington and the Toronto turbines were designed and built by the I. P. Morris Company, Philadelphia, Penn.

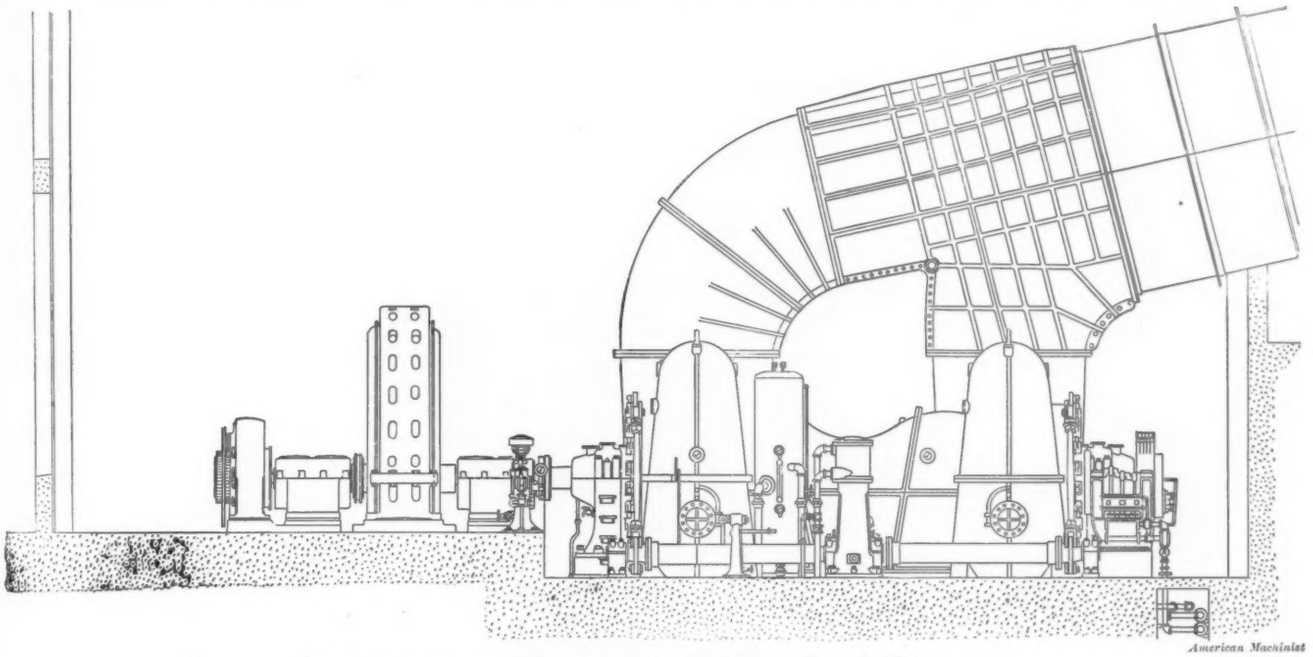
*Assistant hydraulic engineer, I. P. Morris Company.

Power House Arrangement and Type of Turbines

Referring to illustration Fig. 1, the power-house arrangement, it may be seen that the turbines are of the horizontal shaft, double-volute casing, central-discharge type, having two runners mounted on a common shaft, one runner revolving in each volute casing.

Each turbine is supplied with water through a plate-steel penstock 14 feet 10 inches diameter, which is connected to the two volute casings by a heavy cast-iron intake Y-pipe. The Y-pipe, which is located above the turbine, is supported on the two casings and to one side of the vertical plane which passes through the axis of the shaft. The intake Y-pipe is made of cast iron and in nine sections, and it connects the 14-foot 10-inch diameter penstock to the two 8-foot 10-inch diameter intakes of the two volute casings.

To facilitate handling and shipping, the cast-iron volute casings are each made in four sections, the divisions occurring on vertical and horizontal planes. The horizontal joint, which is in a plane passing through the axis of the shaft, measures 24 feet 8 inches across.



American Machinist

Fig. 2. Elevation of Turbine of Little Falls Plant, Washington

Construction

The main turbine shaft is a solid steel forging 36 feet 8¾ inches in length, and passes through both casings and the draft chest, and on its inboard end has a forged coupling which is bolted to a similar coupling on the generator shaft. The diameter of the shaft is increased from both ends to a maximum at its center, so as to eliminate serious deflection at the center. A bearing at the center of the shaft is, therefore, not required.

The two runners are of the inward flow Francis type and each is a single casting of special turbine bronze. Each runner is securely held to a large hub forged on the shaft by means of taper reamed bolts. All vane surfaces are made smooth and the intake edges are sharpened to a knife edge.

The water is distributed to each runner through cast steel movable guide vanes which are located around the periphery of the runner and in the throat of the casing. The vanes turn between two cast-steel removable vane plates. Each vane has cast with it a fulcrum pin, which extends from both sides of the vane. The part of the pin on the draft-chest side of the vane is short in length and extends into a bearing in the cast-steel vane plate. The operating part of the pin, which is long, extends through a stuffing box and two bearings in the head plate. Therefore, each vane pin is held rigidly in alinement by three bearings. The operating side of the pin is connected by a cast-steel lever and a cast-iron link to the cast-steel operating ring, which turns in a groove in the head plate. This arrangement is known as the outside operating mechanism.

The operating ring on each head plate is connected by two reach rods to the main operating shaft, which extends along the side of the turbine and turns in four bearings secured to the floor. A bearing is located at each end of the operating shaft and the two other bearings are located close together, one on either side of the center of the shaft.

A heavy cast-steel lever is secured to the operating shaft between the two center bearings, and to this is attached the piston rod of the operating engine, which is controlled by the governor.

A cast-iron draft chest is located between the two volute casings and is made up of nine sections. It receives the discharge water from both runners and directs it vertically downward into the draft tube. A special cast-iron baffle plate is arranged in the draft chest to direct the water into the draft tube with as little loss as possible.

Between the draft chest and the draft tube is a cast-iron base iron, which is imbedded in the concrete. It is elliptical in shape and measures 12 feet 4 inches on its major axis and 9 feet 3 inches on its minor axis. The lower flange of the draft chest is supported on and securely bolted to the upper face of the ring.

The main turbine shaft revolves in two babbitted, ring-oiling, self-alining generator type bearings, one on either end of the shaft. Each bearing pedestal is mounted on a sub-base which in turn is mounted on a sole plate securely bolted to the concrete foundations. As the two runners discharge in opposite directions, the hydraulic thrust on the shaft is theoretically balanced. Should, however, any unbalanced thrust develop, it will be taken care of by the automatic balancing device arranged with each runner.

The tail end of each turbine is fitted with a mechanical brake, by means of which the revolving parts may be brought to a stop in a few minutes after the gates of the turbine have been shut.

Governors

The speed of the turbines will be controlled by I. P. Morris Company's governors of the double floating-lever, oil-pressure type. Each governor is arranged for belt drive from the switchboard. A friction clutch is furnished with each governor, so that the revolving element may be stopped during the operation of the turbine, without disturbing the driving belt.

The cylinder of the vertical operating engine is fitted with suitable valves and a small hand-operated, oil-pressure pump, so that the supply from the governor may be shut off and the gates of the turbine operated by hand.

The governing apparatus for each of the turbines consists of a governor, operating engine, accumulator tank, three-throw oil pump and the interconnecting piping. The accumulator tank is made of plate steel and is divided into pressure and vacuum compartments. The cubical capacity of the pressure compartment is three times that of the vacuum compartment.

The oil pump is of the Deane three-throw type and is driven from the turbine shaft by means of a belt. The interconnecting piping and fittings between the pump, governor, accumulator tank and operating engine, are made of brass and all the parts are so designed that the governor will control the gates of the turbine satisfactorily with a working pressure not to exceed 150 pounds per square inch. Connections will be made between the governing systems of the four units, so that any pump may supply oil to any accumulator tank.

Each governor is capable of opening or closing the gates of the turbine within two seconds of time and is also capable of being readily adjusted so that the gates may be opened or closed in any given time, up to eight seconds.

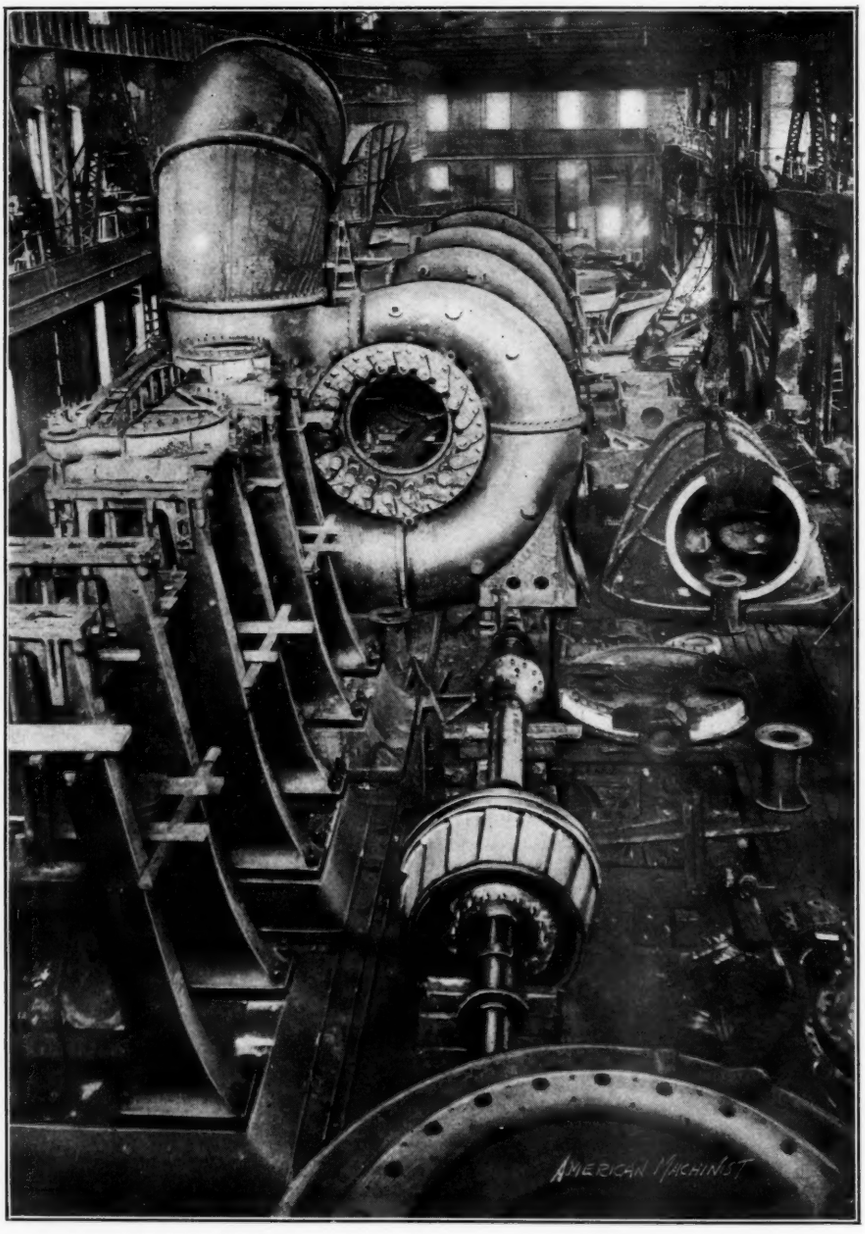


FIG. 3. LITTLE FALLS TURBINE BEING ERECTED IN THE SHOP

In order that the four governors may be operated in parallel, they have an adjustment by which the percentage of inherent drop in speed from no load to full load may be varied from 0 per cent. to 3 per cent. The governors are sensitive to within one-half of 1 per cent., and for a change in speed of one-half of 1 per cent. they will immediately move the gates. On sudden changes in the load, each governor will immediately move the gates of the turbine to the proper position, without excessive hunting.

The weight times the square of the radius of gyration, or the flywheel effect of the generators, is 3,100,000; the length of the penstocks is 150 feet, and the maximum velocity in the penstocks corresponding to full load on the turbines, is 8.7 feet per second. Thus it is seen that the plant conditions are favorable for good regulation.

The facility with which the various parts of turbines are handled in erection or in dismantling for repairs, is a very important consideration in units of this size.

In the case of the Washington turbines, particular attention has been paid to the design of the various parts so that each may be removed without disturbing the parts adjacent to it, any more than is absolutely necessary.

Referring to Fig. 2, the reader will note that the bearings may be readily removed from the shaft without requiring the position of the shaft to be altered. Thus, if it is desired to remove one of the bearings, the bolts connecting the cap to the pedestal are removed and the bolts which connect the pedestal to the soleplate are withdrawn. The sub-base is then slipped out and the pedestal dropped down and removed from beneath the shaft.

Before removing the sub-base of the bearing, it will be necessary to take the weight of the shaft off the bearing, by supporting it at some other place. The means for doing this depends entirely upon the part of the apparatus which is to be dismantled. If it should be desired to take both pedestals from underneath the shaft and dismantle both head plates, the weight of shaft may be taken care of by entering the draft chest and inserting wedges in the clearances between the runner and the draft chest. If it should be desired to remove the tail bearing only, the weight of that end of the shaft may be carried by inserting the wedges in the runner clearances, but if it should be desired to remove the inboard bearing only, the inboard end of the shaft may be carried by the coupling.

If it should be desired to replace a guide vane, it would then be necessary to remove the head plate. The head plate may be removed with everything intact, except the cover plate, which contains the stuffing box of the main shaft. The cover plate is made in two sections, the division occurring on the horizontal plane which passes through the axis of the shaft. The sections are held together by bolts in the flanges on this plane.

The reach rods connecting the operating shaft to the operating ring are first disconnected and the bearing is then dismantled. The nuts on the studs connecting the head plate to the casing are re-

moved, and the nuts connecting the cover plate to the head plate are removed.

The cover plate may then be slipped back out of the way, or by removing the bolts on the horizontal flange, it may be removed from the shaft. The weight of the head plate is then carried by the crane. To make the head plate hang perfectly plumb, a porter bar is fastened to its lower side and enough weights are attached to the end of the porter bar to cause the head plate to hang in a true vertical position.

The head plate may then be backed out of the machined fit in the casing by means of jack screws, and with the crane it may be carried back the distance necessary to allow the removal and replacement of the guide vane. This method of dismantling applies to either end.

The turbines are so designed that the shaft and runners may be removed without disturbing the alinement of the lower sections of the casings, or the intake Y-pipe, or the upper sections of the two casings which support the Y-pipe. In taking out the shaft and runners, the supporting of the shaft, the removal of the bearings and the removal of the head and cover plates is accomplished in the same manner as already described. The brake and the connections to the governor and oil pump are then removed.

The reader will note that the draft chest is made up of seven sections and that three of these sections form a wedge in the center. By this arrangement, the upper sections of the wedge may be readily removed. When this has been done, the remaining upper sections of the draft chest may be unbolted from the wheel casings and from the lower sections of the draft chest. They may then be backed out toward the center and lifted out of place. The next move would be to dismantle the upper sections of the casings, which are connected to the sections supporting the Y-pipe.

The reader will note that the vertical joint between the two upper sections of the casings is made off center. Thus, when the upper sections of the casings which do not support the Y-pipe are removed, the shaft and runners may be readily lifted out.

Illustration Fig. 3 shows two of the units in course of erection in the shops of the I. P. Morris Company, and in the foreground is seen one of the shafts. One of the runners has been forced on the shaft and bolted to the forged hub. The enormous size of these units may be readily appreciated by the reader by referring to this illustration.

Each turbine is guaranteed to have an efficiency of at least 80 per cent. at full load, 82 per cent. at seven-eighths load, and 82 per cent. at three-quarters load; full load being 9000 horsepower, when operating under a head of 66 feet and running at a speed of 150 revolutions per minute.

Engine, Turbine or Combination

The most recent of the torpedo-boat destroyers acquired by the French government are divided into three types with respect to the engines used. The "Chasseur" and the "Actèe" are propelled by three sets of turbines each. Those of he former are of Parsons design and those of the latter are of the Schneider-Zoelly type. The "Voltigeur" and "Tirailleur" each have one set of triple-expansion, three-cylinder engines and two sets of turbines. The turbines in the "Tirailleur" are of the Breguet-de Laval type while those in "Voltigeur" are of Rateau build. The "Spahi," the "Carabinier" and the "Hussard" are fitted with triple-expansion, reciprocating engines; those of the first two boats have four cylinders while those of the last named have three cylinders.

A comparison of the performance under test of boats representing each type is offered in the accompanying diagrams and table which are taken from a bulletin of the Société des Anciens Elèves de l'Ecole Professionnel e de Nantes.

In the combined diagram, Fig. 2, the cross-hatched portion represents the work done by the low-pressure turbine with steam, after a triple-expansion engine has got through with it.

The results are plotted with the speeds in knots as the abscissas and the coal consumption for the ordinates. The consumption per hour of the "Carabinier" is less than that of either of the turbine boats at speeds up to about 25 knots. Above this it becomes superior to that of either of the turbine boats and is rapidly rising. The displacement of the "Carabinier"

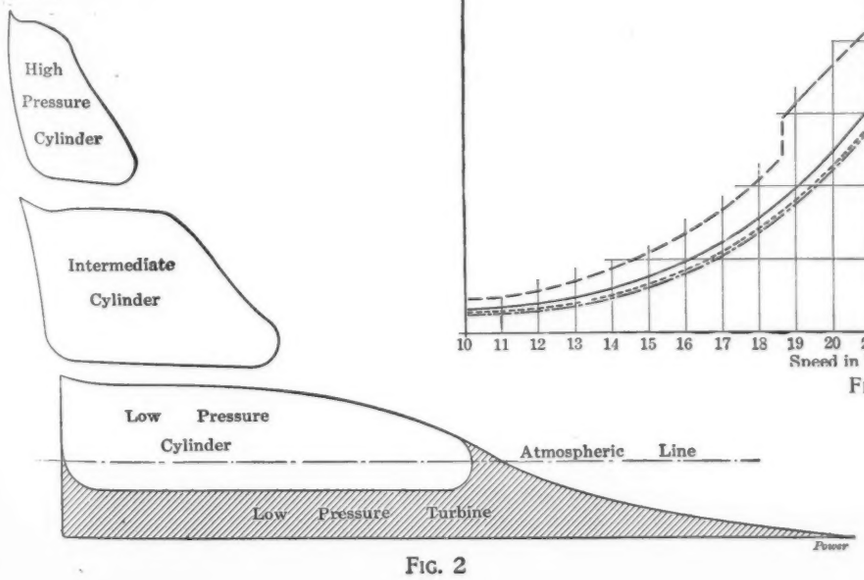


FIG. 2

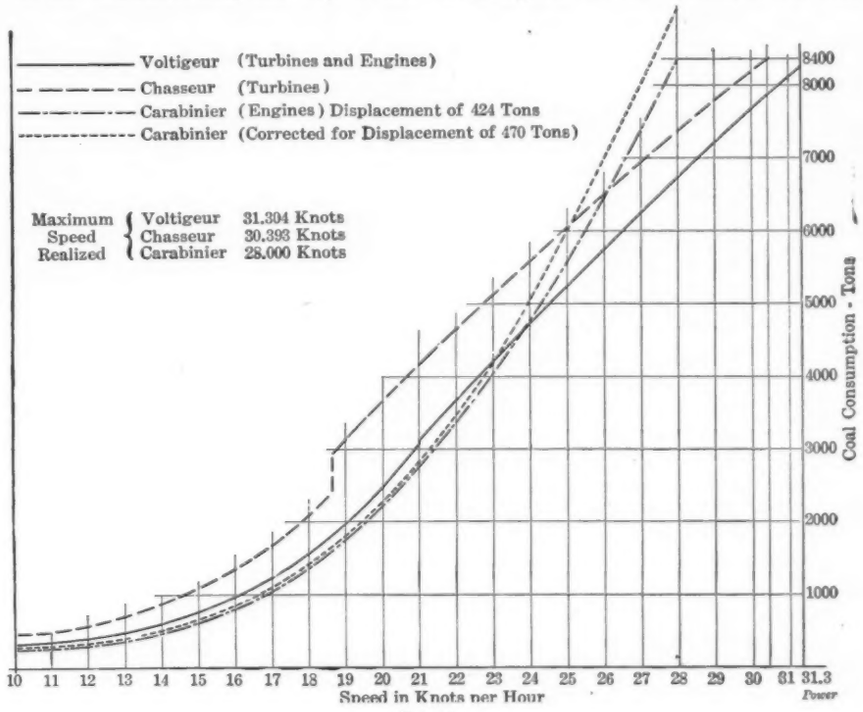


FIG. 1

COMPARATIVE PERFORMANCE OF RECIPROCATING ENGINES AND TURBINES.

	Chasseur.	Voltigeur.	Carabinier.
Engines	3 Parsons turbines; 1 high-pres., 2 low-pres. 2400 h.p. at 850 r.p.m.	Triple expansion engine; diam. 440-635-956 mm. Stroke 500 mm. R.p.m., 380. 2 Rateau low pres. turbines, 3000 h.p. each, at 850 r.p.m.	2 reciprocating engines; dimensions not given.
Trial for Eight Hours at 14 Knots.			
Average speed	13.872 knots	14.09 knots	14.023 knots
Displacement during test	470 tonnes	468 tonnes	424 tonnes
Coal consumed: per hour	965 kilos	636 kilos	455 kilos
Coal consumed: per sq. meter of grate per hr.	89 kilos	58.636 kilos	42.523 kilos
Coal consumed: per mile at 14 knots	69.56 kilos	45.191 kilos	32.339 kilos
Coal consumed: to go 1170 miles	81,000 kilos	52,873 kilos	37,836 kilos
Trial for One Hour at Maximum Speed.			
Average speed	30.393 knots	31.304 knots	27.7 knots
Coal consumed: per hour	8,480 kilos	8,280 kilos	8,400 kilos
Coal consumed: per sq. meter of grate per hr.	392.6	376.36	392 kilos
Coal consumed: per mile	879	264	303 kilos
Coal consumed: per mile at 30 knots	274	258	
Trial for Five Hours at 24 Knots.			
Average speed	25.56 knots	24.74 knots	24 knots
Coal consumed: per hour	6,448 kilos	5,112 kilos	4,800 kilos
Coal consumed: per sq. meter of grate per hr.	2 98 kilos	232.363 kilos	224 kilos
Coal consumed: per mile	252 kilos	207 kilos	200 kilos
Coal consumed: per mile at 24 knots	232 kilos	198 kilos	200 kilos

is 46 tons less than that of either of the other two boats, and the dotted line is based upon the estimated consumption corrected for this difference. In the "Chasseur" the consumption from 10 to 26 knots is superior to that of the reciprocating engines; on the contrary, it becomes notably less from 26 to 31 knots.

On the "Voltigeur" the consumption at slow speeds, from 10 to 24 knots, is very little more than that of the reciprocating engines, and considerably less than that of the all-turbine boat, due to the use of its reciprocating engine exhausting to the turbines or to its special condenser, according to the speed and power required. At high speed the economy realized is remarkable. For the same quantity of carbon burned during the test of one hour at high speed the "Voltigeur" attained the speed of 31.3 knots—greater by nearly three knots than that of the "Carabinier," with the reciprocating machines, and by 0.9 of a knot than that of the "Chasseur," with the Parsons turbines. A comparative trial between the "Chasseur" and the "Voltigeur" was made to determine the distance in which they could be stopped. When both vessels were going at a speed of 20 knots the word was given to reverse the turbines, and after the word was given the "Chasseur" went to 200 meters and the "Voltigeur" 167. The boats with the reciprocating engines went from 150 to 180 meters before coming to a stop.

Common Diseases of Boilers

Lap Seam Cracks, Closing up of Joints, Loss of Ductility, Defective Bolts, Incrustation and Scale, Oil, Corrosion and Leaking at the Seams

Recently, in an address to the New England Association of Electric Lighting Engineers, F. S. Allen, chief inspector of the Hartford Steam Boiler Inspection and Insurance Company, discussed the diseases to which steam boilers are subject.

One of the chief manifestations of trouble is the development of cracks at the lap seam along a line which passes close to the rivet holes but usually does not enter them except when radiating branch cracks are present. These lap-seam cracks may be caused by a number of conditions. They are more prevalent in the modern boilers, in the construction of which steel is used, than in the older types which were built of iron. This would indicate that the quality of the material is an influencing factor. A circumstance which has an influence also is that iron could not be obtained in large plates; thus the distance between the girth seams was less in the boilers of the older type.

Steel appears to be more sensitive than iron to the slight local movements which occur near the joints due to the variations of pressure in the boiler and to the fact that the contour of the shell is not truly circular near the joints. Often, too, the plates are not bent in the rolls so that they conform to each other at the end. This causes a closing up of the joints as the rivets are driven which results in a severe permanent strain in the material which doubtlessly hastens the formation of a lap-joint crack.

Defective Tubes

Boiler tubes have been found to lose their ductility after having been in service for but a few years. Undoubtedly the skelp from which such tubes are made is of an inferior quality, and the alteration in their structure, with the resulting liability of fracture, is probably due to the temperature to which they are exposed. There is much difficulty in detecting alteration in the structure of a boiler tube, although there is not infrequently some unusual color to the tube or some unusual sound given out under the hammer test that will attract the attention of the examiner.

Defective Bolts

Of late many defective bolts have been found by the Hartford company's inspectors among those that are used for holding the tube caps onto the manifolds in some types of water-tube boilers. This is a dangerous defect in Mr. Allen's opinion, as most of these boilers are operated under high pressure, and the caps are upon the outside so that the failure of the bolts releases the whole contents of the boiler into the fire room. The failure of the bolts is caused by loss of ductility. Mr. Allen believes that the remedy is to use bolts forged from the very best quality of Swedish iron.

Effects of Scale

Another cause of rapid deterioration and loss of efficiency in boilers is the formation of incrustation and scale. Water-tube boilers are peculiarly sensitive to this, as their tubes are liable to become overheated and the thin material of which they are made then becomes subject to distortion, where the relatively heavy plates of a shell boiler would remain comparatively unaffected. A great many cases of this kind occur yearly, and the rupture of a tube is not an infrequent occurrence. Rapid deterioration of the tubes may be caused by the feed water, and the nature of the incrustation it forms. The failure of tubes in water-tube boilers is sometimes due to defects in construction or in welding.

The increase in temperature attendant

upon the use of higher pressure has brought about some new developments detrimental to boilers in connection with the formation of scale, especially in the fire-box type of upright boilers.

In a certain case, trouble occurred in a battery of large boilers of this upright type, operating at a pressure of 170 pounds per square inch. In the same plant were a number of older boilers of the same type, operating at 125 pounds. There had never been any trouble from scale, although in the older boilers there had been a considerable deposit of mud which could readily be removed by periodical washings. No trouble from leakage had been experienced from this sediment at any time until the new high-pressure boilers were installed for electric power. An examination of the high-pressure boilers after the leakage around the tube ends had developed showed a thin, hard coating of sulphate of lime over the whole tube sheet, making a slight fillet around each tube. The coating resembled an enameled lining more than scale, owing to its extreme thinness and its adherence to the plate. The fact that the same water had been used in the boilers in operation in the plant for over 20 years and that no trouble had occurred from scale or deposit, made it difficult to persuade the engineer that the leakage was due to the feed water and the scale formation; but by the judicious use of solvents the enamel-like coating was finally dissolved and no leakage has occurred since, solvents being now used to prevent further deposits of scale.

Oil in Boilers

Oil continues to be a great source of injury and destruction, where the water of condensation is recovered from the exhaust steam, and used over again in the boilers. There seems to be much difficulty experienced in removing oil from boilers, when it has once effected an entrance. This can be done readily, in some types of boilers, by swabbing the sheets and tubes with a mop dipped in kerosene oil, after taking the highly important precaution of extinguishing all open lights about the boiler, as a measure of safety. In other cases, where the boilers are inaccessible for mopping, they can be boiled out with a strong solution of soda ash (or caustic soda if the soda ash does not prove effective), with a generous addition of kerosene oil, the pressure being maintained at half or two-thirds of the regular working pressure for 12 to 20 hours.

Corrosion

Corrosion is not so common today as formerly, but it still is an active enemy of steam boilers. It is less common because a great change has been made in the last few years in the types of boiler in general use, and those that are at present most common are less liable to corrosive action than were the drop-flue, hammer-head, and similar types having a poor circulation at the bottom. Each case must be carefully investigated, and a remedy applied that is appropriate to the cause of the difficulty. Where the water is pure and the boilers are operated intermittently, corrosion is frequently found in the form of pitting. This action takes place very often in pumping stations, and in power plants and electric stations where the fires are kept banked for long periods.

Once started, corrosion is likely to go on until the material of the boiler is destroyed, unless measures are taken to check it. When corrosion is observed in connection with the use of a pure water, one of the best methods of treatment, Mr. Allen said, is to keep the water alkaline with soda ash, for this tends to check the corrosive action, and the soda does not injure the boiler.

Nitrates in Feed Water

Care should be exercised in selecting feed water for a new plant, or for a new location of a plant, to see that the quality of the water is good. Nitrates in the water should be especially avoided, as they are especially troublesome and dangerous.

Influence of Operating Conditions

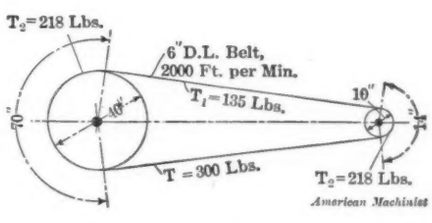
Many of the minor diseases of boilers, such as rapid loss of ductility, and development of incipient fractures at different parts, as at the girth joints in the plain tubular boiler, may be due to the conditions under which the boilers are operated, such as a varying level of the water, the introduction of cold feed water, or blowing down the boilers under high pressure and leaving the drafts on, so that cold air may be drawn through and give rise to serious unequal contraction, or pushing the fires too hard in raising steam from cold water. Severe strains, resulting in leakage at the seams and around the staybolts and tube ends of fire-box boilers, are frequently caused by the burning out of the fire under one boiler of a battery, while this boiler is left connected with the rest of the battery, and with the draft on full.

Power Loss of a Slipping Belt

One often sees discussed the question whether or not a slipping belt is less economical of power than one which does not slip. If we consider that when a belt slips friction occurs between the pulley and belt, thus creating heat, it is at once seen that part of the driving power is "lost," as the radiation of this heat attains no useful end. The approximate amount of this loss may be obtained as follows:

A motor delivers ten horsepower to a line shaft by means of a 6-inch double leather belt running 2000 feet per minute. The accompanying sketch shows the belt tension T to be 300 pounds on the tight side, while the tension T_1 on the slack side is 135 pounds. The average belt tension T_2 will then be about the half sum of the other two, or 218 pounds.

The tension in the walls of a thin cylinder due to a uniform pressure is equal to the product of the unit pressure multiplied by the radius of the cylinder. Conversely, since the average belt tension at the pulleys is 218 pounds, the average belt pressure per inch length of contact with either pulley is equal to 218 pounds divided by its radius in inches. This amounts to 11 pounds for the large pulley and 44 pounds for the small one.



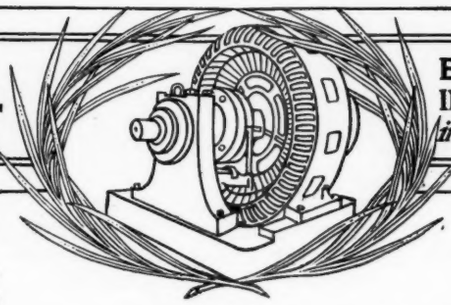
Tensions in the Belt

Since the lengths of belt contacts are 70 inches and 14 inches respectively, the total pressure of the belt on the large pulley is 770 pounds, and on the other 616 pounds.

If the total belt slip be 10 per cent. the slip on the motor pulley will be about 6 per cent., or inversely proportional to the total belt pressures. A 6 per cent. slip on the motor pulley amounts to 120 feet per minute. If the coefficient of friction of leather on iron is 0.2 then the grip of the belt on this pulley amounts to 616x0.2, or 123 pounds. The loss of work is then 123 pounds multiplied by 120 feet per minute or 14,760 foot pounds per minute. In like manner, the belt slip on the large pulley being 4 per cent., the pulley rim travels 80 feet per minute slower than the belt. The grip of the belt on this pulley is 154 pounds and the energy lost per minute is 12,320 foot-pounds.

The total energy lost in this drive due to a 10 per cent. belt slip is then about 27,000 foot-pounds per minute, or seven-eighths of a horsepower. In the above computations centrifugal force on the belt has been neglected.

ELECTRICAL DEPARTMENT



ESPECIALLY CONDUCTED *to* BE *of* INTEREST *and* SERVICE *to the* MEN *in* CHARGE *of the* ELECTRICAL EQUIPMENT

Primer of Electricity

In previous primer lessons it was explained that the motion of a wire across a magnetic field induces an electromotive force in the wire. This principle holds good, of course, for any conductor of electricity, whether in the form of a wire or a rod or an irregular "chunk." Therefore, if the armature core shown in Fig. 44 were a solid cylinder of iron, as the sketch appears to indicate, revolving it between the poles of a magnet in

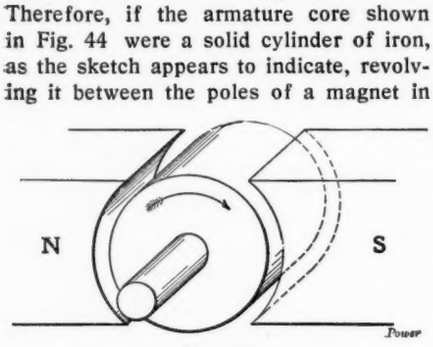


FIG. 44

the direction of the arrow would cause currents to flow from front to back in that part of the core passing under the "north" pole of the magnet, and from back to front along the part passing under the opposite magnet pole, these currents completing their circuits across the heads of the core, as indicated by the arrows in Fig. 45.

Of course, this circulation of currents in the core would waste just so much energy, which would manifest itself by heating the core. Therefore, it was found necessary, very early in the development of the dynamo, to do something

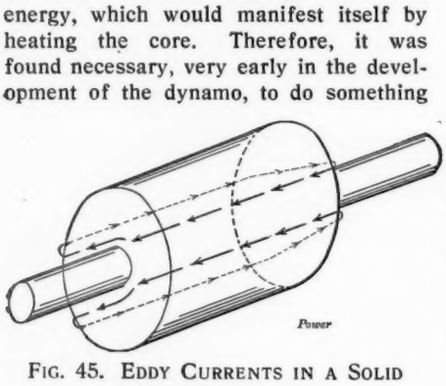


FIG. 45. EDDY CURRENTS IN A SOLID ARMATURE CORE

to prevent such currents from being created. It was impractical to make the core of nonconducting material, because it must be of a magnetic material in order to carry the flux across from pole to pole efficiently, and there is no magnetic material that will not conduct electricity also.

Consequently, the plan was adopted of dividing the core into "layers" and insulating the layers from each other so the undesired currents could not get across from one layer to the next.

Fig. 46 shows such a core, partly in section in order to illustrate the construction. Instead of being a solid cylinder of iron, the core is built up of a lot of thin iron disks threaded on the shaft and insulated from each other either by a coat of varnish or a very thin layer of paper on each disk.

Induced currents always flow at right angles to the magnetic flux which induces them and to the direction of motion by the conductor in which they are induced. Therefore, in an armature core they tend to flow parallel to the shaft, and as the metallic path through the core is broken by the insulation between the disks, there is not the circuit for them around the core that there would be in a solid core.

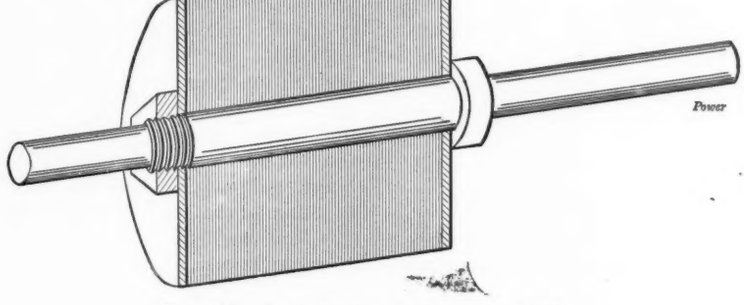


FIG. 46. LAMINATED ARMATURE CORE

In spite of this precaution, however, currents are induced in each individual disk, passing across the edge, inwardly across the central part to the other edge, across that edge of the disk and back to the edge first mentioned, as represented by the arrows in Fig. 47, where the thickness of the disk is exaggerated to make the effect clearer.

These objectionable currents are called "eddy" currents, because of their eddy-like course through the disks. Some of them do not travel all the way across the diameter of the disk, but cut through the metal near the shaft and return to the edge along the other face of the disk, as indicated at *a* and *b*.

The thicker the disks, the greater will be the flow of eddy currents, because the resistance of their individual circuits will be reduced; therefore, the disks used in practice are as thin as they can be made without increasing the cost of the core too much. Beyond a certain point, the increase in the cost of the core would be more than the reduction in energy loss by eddy currents would justify. In most armatures, the disks are from 14 to 25 mils thick, according to the kind of the machine in which the armature works. (A mil is one-thousandth of an inch.)

The extent to which eddy currents are created in an armature core depends upon four features of construction and operation: the thickness of the disks, the electrical conductivity of the iron or steel of which the disks are made, the density of the magnetic flux in the disks and the speed at which they revolve.

The reasons why these points all affect the extent of eddy-current loss are as follows: The e.m.f. induced in the disk by its motion in the magnetic field depends entirely upon the strength of the magnetic field (flux density) and the speed at which the disk "cuts" that field; eddy currents, like all others, are proportional to the e.m.f. which produces them and the conductivity of the circuit in which they flow; the conductivity of a disk depends on its thickness (because that determines the cross-sectional area of the path) and the specific conductivity of the material.

Without going into the mathematical theory any farther, it is sufficient to say that the eddy-current loss, in watts, is proportional to

$$\textit{No. magnet poles}^2 \times \textit{r.p.m.}^2$$

and to

$$\textit{Magnetic density}^2$$

and to

$$\textit{Disk thickness}^2$$

and to

Cubic inches in the core.

These relations are shown by the five curves in Fig. 48, which are plotted for a good grade of sheet steel 14 mils and 25 mils thick, at 100 revolutions per minute, between two magnet poles.

The chart may seem a little compli-

cated at first, but its application is simple. The upper portion of it relates to the features of magnetic density and thickness of metal, the scale on the left being densities and the two curves representing the two thicknesses of sheet steel commonly used for armature cores.

The lower part of the chart takes care of the number of magnet poles between which the armature revolves.

Suppose you had a four-pole dynamo with an armature 24 inches in diameter, 10 inches long and having a 12-inch hole through the center, running at 600 revolutions per minute; also suppose that the magnetic flux from each magnet pole was 7,560,000 lines of force, and the steel disks were 25 mils thick. The net quantity of steel lengthwise of the armature would be about 9 inches, because of the space lost between disks. The volume of steel in the core would be 3054 cubic inches and the magnetic density 70,000 lines per square inch.

Now, referring to the upper part of the diagram, start at 70,000, in the density scale, follow that horizontal line to the 25-mil curve, which you will strike just half-way between the two vertical lines corresponding to 1.2 and

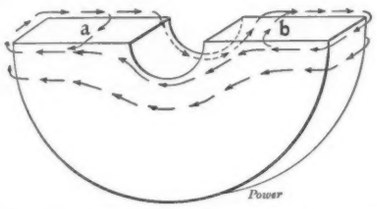


FIG. 47. EDDY CURRENTS IN A DISK

1.3 on the top scale. Trace downward, midway between these two lines until you strike the diagonal line marked "4 poles"; from this point trace the horizontal line over to the left-hand edge of the chart, where you will find the number 14.

This means that 14 watts will be wasted by eddy currents in each 10,000 cubic inches of core at 100 revolutions per minute. But there are not 10,000 cubic inches in the core—only 3054; therefore the loss at 100 revolutions per minute would be

$$\frac{3054}{10{,}000} \times 14 = 4.2756$$

watts.

Since the loss is proportional to the square of the speed ($r.p.m.^2$) and the speed is 600 revolutions per minute instead of 100, or six times the speed at which 4.2756 watts would be lost, the loss at 100 revolutions per minute must be multiplied by $6^2 = 36$, in order to ascertain the loss at the actual speed of the armature. The actual loss will be, therefore,

$$4.2756 \times 36 = 153.92$$

watts—practically 154 watts—for the actual core at the actual speed.

Now, 154 watts is one-fifth of a horsepower, but a 24-inch armature core is a pretty large one, so the relation of the eddy current loss to the output of the machine would not be very great.

The chart shows clearly the effect of magnetic density, disk thickness and number of magnet poles. Take, for example, an armature containing just 10,000 cubic inches of steel, and running at 1000 revolutions per minute, to simplify

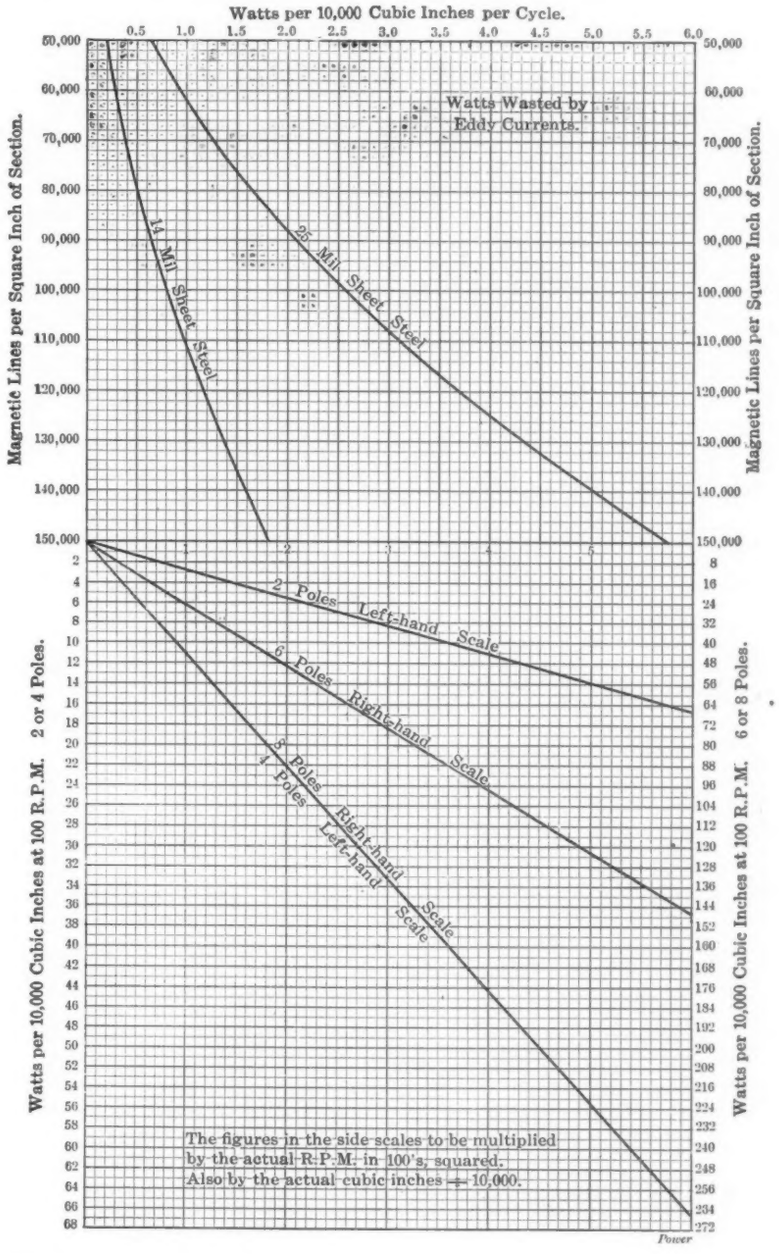

Watts per 10,000 Cubic Inches per Cycle.
0.5 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0 5.5 6.0
Magnetic Lines per Square Inch of Section.
Watts Wasted by Eddy Currents.
14 Mil Sheet Steel
25 Mil Sheet Steel
2 Poles Left-hand Scale
6 Poles Right-hand Scale
8 Poles Right-hand Scale
4 Poles Left-hand Scale
Watts per 10,000 Cubic Inches at 100 R.P.M. 2 or 4 Poles.
Watts per 10,000 Cubic Inches at 100 R.P.M. 6 or 8 Poles.
The figures in the side scales to be multiplied by the actual R.P.M. in 100's, squared. Also by the actual cubic inches ÷ 10,000.
Power

FIG. 48. CHART SHOWING WATTS WASTED BY EDDY CURRENTS IN ARMATURE CORES

lutions per minute), whereas if 14-mil disks were used the loss would be reduced to 160 watts.

It must not be supposed that all this discussion of eddy currents is "high-brow theory," useful only to the student who wishes to learn how to design electrical machinery. It is of much importance to the practical operating engineer to understand eddy currents, as well as other things which may affect the efficient working of machines in his charge: what causes them, what their effects are, and how to prevent them from increasing.

For the present, the practical bearing of eddy currents will not be gone into, because other sources of heating and losses should be considered first. The next primer lesson will explain another cause of heating and loss in armature cores.

the case. With a density of 52,000 in 25-mil disks, the loss with two poles would be 200 watts; four poles, 800 watts; six poles, 1800 watts; eight poles, 3200 watts.

Again, with 52,000 density in 25-mil disks and two poles the loss would be 200 watts, while at 95,000 density the loss would be 625 watts.

Once more, with two poles and a density of 84,000 lines in 25-mil disks the loss would be 500 watts (at 1000 revo-

Inexpensive Alternating Current Station Recording Watt Hour Meters

By H. R. Mason

In order to secure the best results from an electrical generating station, it is absolutely necessary to measure the output accurately; then it will be possible to determine whether the costs are excessive. There are many small stations about the country which have neglected this feature, evidently assuming that the cost of installation would not be justified. It is true that the expensive, highly finished recording wattmeters usually mounted on the switchboard may be too costly, but it does not seem to be generally recognized that the ordinary type of service meter may be employed in most alternating-current stations, with excellent results, by using suitable transformers and paying proper attention to the connections.

The potential transformer must deliver a voltage suitable for the meter which is to be used, and the current transformer must be so proportioned that the current in the meter will not be excessive when full load exists in the circuit which is to be metered. Potential transformers are very small transformers which deliver a secondary voltage in direct proportion to the line voltage, like the ordinary types used for lighting and power service; a current transformer, however, delivers current at its secondary terminals which is proportional to the current flowing in its primary circuit.

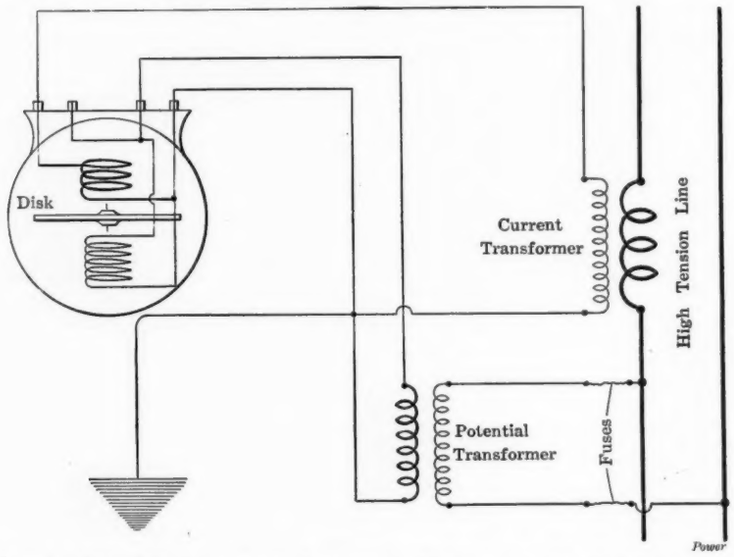


Fig. 1. Connections of an Induction Meter and its Transformers

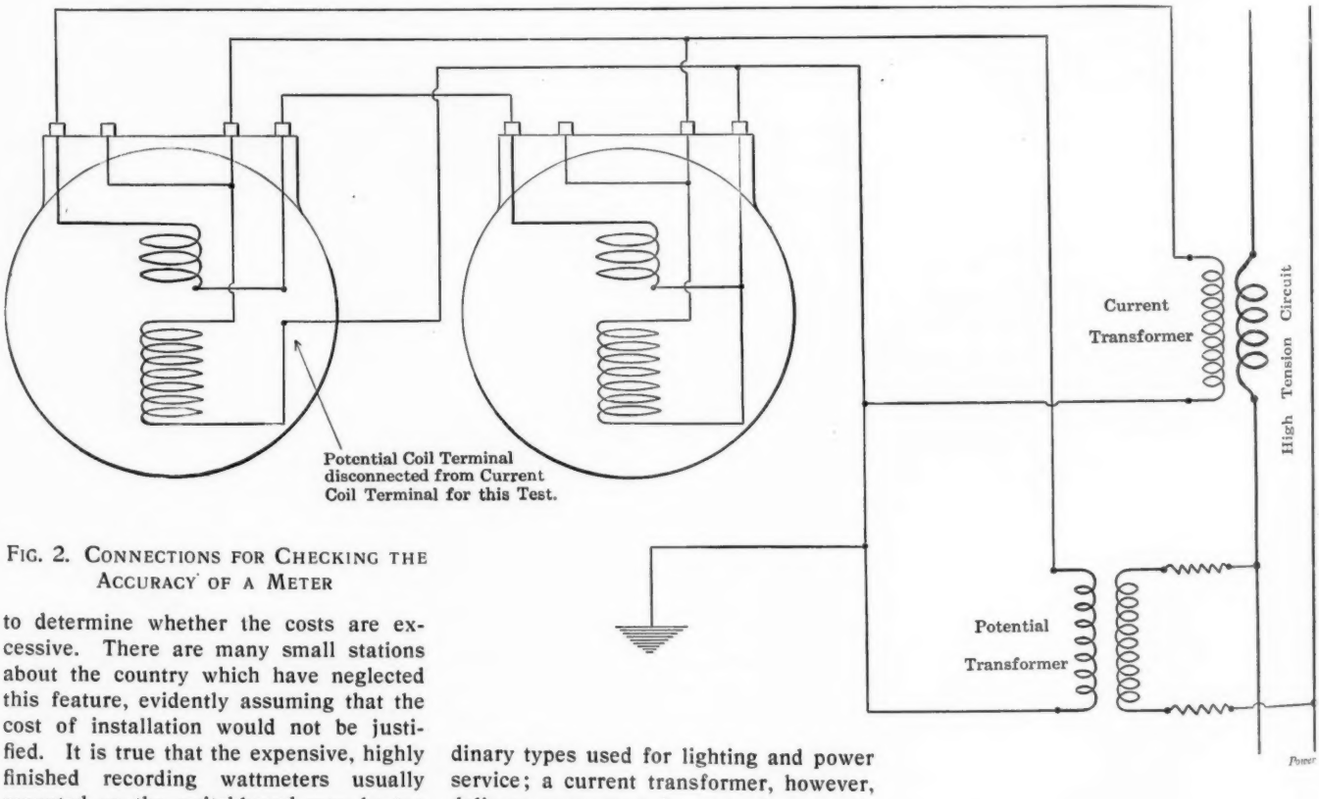


Fig. 2. Connections for Checking the Accuracy of a Meter

The secondary winding of a current transformer is intended to deliver current to a circuit of very low resistance, which would almost instantly blow the fuses of the ordinary type of transformer; while in normal operation there is an extremely small difference of potential between its terminals. The secondary circuit of a current transformer should be closed through a conductor of very low resistance in case it becomes necessary to disconnect instruments from the transformer if the circuit to which the transformer primary is connected remains in operation, and some of the standard makes are provided with special terminals to facilitate this operation. The primary winding of the transformer, which carries the entire current flowing through the circuit to be measured, consists of a very few turns of heavy wire in the smaller transformers, and a single turn in the larger sizes; the secondary winding contains a number of turns proportional to the reduction in current strength desired, usually ranging from 5 to 100 times the number of turns in the primary.

In operation, the current flowing through the primary coil sets up a magnetic flux in the iron core of the transformer, which in turn causes current to flow through the secondary winding, the secondary current being such that the ampere-turns of the primary and secondary coils are equal. For example, if a current transformer having 5 turns in the primary coil and 200 in the secondary should have 100 amperes flowing through the primary, a current of 2½ amperes would flow through the secondary winding if the resistance were not excessive; the ampere-turns in the primary coil would be 100 amperes × 5 turns = 500, while the ampere-turns in

the secondary would be 2½ amperes × 200 turns = 500, also.

Resistance in the secondary circuit will not materially alter the amount of current which will flow unless allowed to become excessive; it will only result in increasing the magnetic density in the core of the transformer and creating a considerable difference of potential between the terminals. This may be carried to a highly objectionable degree, as it may cause a very undesirable loss of energy and reduction of the line voltage due to the "choking" effect of the transformer under such circumstances. If the secondary circuit is opened this condition reaches its worst, and full-load current flowing through the primary coil will cause an exceedingly high magnetic density in the core, with a correspondingly high difference of potential between the terminals of both the primary and secondary windings, which will almost invariably result in the destruction of the transformer, either by overheating of the core or failure of the insulation, and at the same time will increase the voltage drop in the line undesirably; hence the necessity for keeping the secondary circuit always closed while current is flowing through the primary winding.

The internal connections of a standard make of the ordinary induction type of service wattmeter are shown at the left of the diagram in Fig. 1, connected to current and potential transformers on a single-phase circuit at the right. The secondary terminals of the current transformer must be so connected that the current flows through the current coils of the meter, which can readily be located upon removal of the cover, as they are composed of comparatively coarse wire; the secondary terminals of the potential transformer must be connected to the potential coils, which are wound with very fine wire. A single return wire for both the current and potential transformer secondaries may be used to advantage, and this wire should be grounded to insure against accident to persons touching the meter if the insulation should become defective in either of the transformers.

The matter of transformer ratios must be carefully considered. The choice of potential transformers is dependent only upon the ratio between the line voltage and that of the meter to be used, but in measuring the output of a station having a very small day load and a heavy evening peak, the ratio of the current transformer must be such that the meter will not be overloaded when the maximum load exists, and yet will be able to furnish an appreciable amount of current to the meter during the hours of light load.

As an illustration, a certain small station generating alternating current at 2200 volts furnished only 20 to 25 amperes during the day, but the load rose as high as 75 amperes in the evening. A potential transformer having a ratio of 20 to 1 was used, and this delivered 110 volts to the meter. To accommodate expected growth, a current transformer having a ratio of 100 to 5 was used, giving a secondary current of 5 amperes when 100 amperes passed through the primary. These were connected to a 5-ampere recording watt-hour meter as shown in Fig. 1. On this type of meter in ordinary service, the extreme right-hand dial indicates tenths of a kilowatt-hour. To determine the amount of energy delivered through the primary circuit, the reading shown by the meter was multiplied first by twenty, as the voltage at the meter was one-twentieth that of the main circuit, and this result also multiplied by twenty, as the current transformer ratio was 20 to 1, so that the current through the meter was one-twentieth that delivered to the lines; this second result was divided by ten, as the meter indicated in tenths of a kilowatt-hour. In practice, this resolved itself into the use of a "constant" of 40; that is, the number of kilowatt-hours shown on the meter dials had to be multiplied by 40 in order to ascertain the actual kilowatt-hours delivered to the line.

Both potential and current transformers are manufactured in a large number of standard ratios, so that almost any set of conditions may be met; the constant by which the meter readings must be multiplied may be determined as just explained. Should the meter disk revolve in the wrong direction upon the first trial, the terminals of either the potential or the current transformer may be reversed, but not both. This will cause rotation in the proper direction.

Meters installed as described may be tested in a number of ways. The current through the current coils may be measured by an ammeter and the voltage delivered by the potential transformer ascertained, and their product multiplied by the power factor of the circuit; but as it is seldom practicable to determine the power factor on installations of this nature, this method cannot usually be applied. The simplest way is to remove the meter and connect it in circuit with a bank of lamps or other suitable load, and compare its indications with a standard test meter in the same circuit, if one is obtainable; if not, a new meter direct from the factory may usually be assumed to be correct, and this may be used as a standard.

The two meters should be connected as shown in Fig. 2, the current coils being in series and the potential coils in parallel. The revolutions of the disks may then be compared, or the meters may be left connected for some time and the readings compared; but if this method is used care must be observed to secure a meter of exactly the same make and type as the one to be tested, or to make due allowance for the difference between the two. If any wide variation is found, it is well to try a second new meter for comparison, as there is always a possibility that the test meter used may have been damaged in transit, and it is very important that no serious error be made in measuring the output of a station.

Though direct-current watt-hour meters operate on somewhat similar principles, having independent voltage and current elements, this system cannot well be applied to them, for in order to obtain a supply for the current coils, a shunt depending upon resistance instead of a transformer depending upon induction would have to be used, and several vital objections would arise. The shunt would need to be of considerable resistance in order to divert sufficient current for the operation of the meter, and would become hot when carrying heavy loads, thus causing a change in its resistance and giving inaccurate results, beside wasting energy. Then, too, the design and construction of such a shunt would involve a vast amount of careful testing and special work, and would be more expensive in the end than the purchase of a suitable direct-current recording watt-hour meter which would give results that could be depended upon.

CORRESPONDENCE

Repairing Holes in Commutators

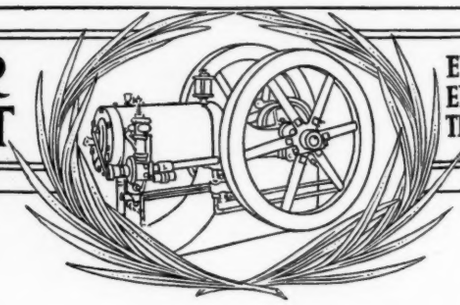
I was much interested in the recent article by G. E. Miles on repairing holes in commutators, and can indorse all he says about the use of litharge and glycerin for that purpose; I used it with success many years ago. However, I have had even better results with powdered asbestos mixed with silicate of soda to the consistency of a paste. If this is allowed to stand for a few hours it will become very hard, and it is very good for unusually large holes in commutators subjected to severe service. I find it to be the best thing I have used yet.

Last fall lightning burned out 30 coils in a 250-kilowatt armature; the commutator was very badly burned and had three large holes in it. At first I thought I would renew the segments, but finally decided to plug it, as it was worth the trial. This generator has run day and night since last September, with a shutdown for about an hour every Saturday afternoon for inspection and cleaning, carrying a load of rolling-mill cranes and open-hearth charging machines; up to this time it has caused no trouble and runs just as good as it did when new.

H. H. RYDE.

Monessen, Penn.

GAS POWER DEPARTMENT



EVERY THING WORTH WHILE *in the* GAS ENGINE *and* PRODUCER INDUSTRY WILL BE TREATED HERE *for the* USE *of* PRACTICAL MEN

Gas Engine Ignition

By Wesley E. McArdell

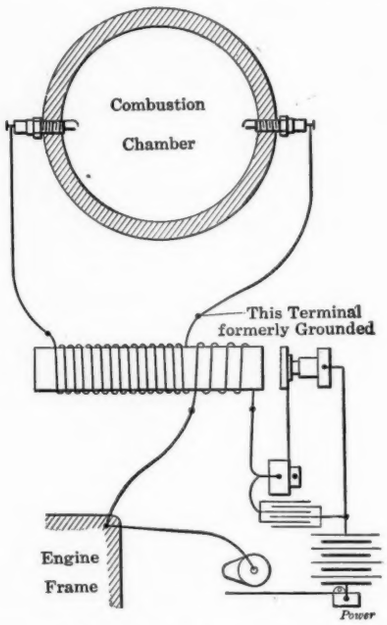
When our engine was operated on city gas we had no trouble with our low-tension system of make-and-break ignition; we renewed the elements in the six Edison-Lalande cells every six months, cleaned the contact points on the plug once a week, and all went along merrily as it should. One day (one month rather), we decided to change over to producer gas. A new cylinder head having a smaller compression space was put on, but the old ignition device was retained, the inference being that having operated successfully on city gas, it would be equally good with producer gas, but we were to learn several things.

The engine would run for a period of time and then miss explosions, and on shutting down we would find embedded in one of the igniter electrodes a minute patch of gritty substance which prevented the circuit from being completed. On examination, the substance proved to be either a piece of coke from the scrubber or a particle of tarry carbon detached from the supply pipe. The remedy was of course obvious, and we bought two of the best spark coils that could be had for money, with an assortment of spark plugs, and tapped out a new cover plate to fit over the hole formerly occupied by the low-tension igniter.

We were somewhat uneasy as to what tendency the old igniter-plug hole would show, by retaining some of the burned gases, toward causing pre-ignition, but we subsequently learned that the action of this space was precisely the same as that claimed by some of the spark-plug manufacturers for their plugs—it shot a veritable blast of flame into the cylinder. The pocket measured 6 inches in length by 2 inches in diameter.

Curiously enough we found that while the old make-and-break plug tripped 20 degrees before the back center was reached, the best results on the new system were obtained by advancing the ignition still farther—to 35 degrees ahead of the center. The engine, too, seemed to run stronger, as though the ignition were more instantaneous.

We learned, however, that even high-tension plugs fail at times and although we got along very well by discovering that it was possible to change the defective plug for a new one by disconnecting the secondary lead and holding open the exhaust valve, the engine having sufficient momentum to run the brief thirty seconds required to make the change, it was at an electrical show that we gained an idea that set us on the right track. They called it a "spark-gap," and that was about all it amounted to. After reasoning awhile we concluded that if we were to have two sparks we might as well have them both in the cylinder, so provision was made for another plug in the side of the compression space and



Circuit Connections with Twin Spark Plugs

the other high-tension lead that formerly was grounded was hooked to its insulated terminal. This arrangement is illustrated by the accompanying sketch, which is, however, only schematic. The windings of the spark coil, for example, are shown side by side, but in the actual coil one is wound over the other, of course; and the timer, vibrator and plugs are merely representative, not accurately pictorial.

By having one spare plug and alternately cleaning the plugs in the engine we had practically no trouble from lack of ignition. We had, it is true, drifted away somewhat from the real purpose of an auxiliary spark gap (of providing an absolute gap and not a partially sooted-over plug) but we had the advantage of having two plugs and if one short-circuited, the other was there to keep up the good work.

Once again the timing had to be changed. This time we had to retard the spark some 17 degrees from the last timing, bringing it back to about 18 degrees, owing to the ignition occurring at two points. We then got a sure sign of a short-circuited plug in the lack of power in explosions.

The advantages of the jump-spark system over the make-and-break ignition are easy removal of the plug, as against the loosening of two or more half-inch studs, making it easy to change plugs without stopping the engine; a steady stream of sparks lasting a quarter or more of a revolution, as against the single flash of the make-and-break system; plugs can be secured that will spark under water, whereas dampness will prevent the working of practically every low-tension igniter the writer has seen.

Three-years' Record From a Diesel Engine Plant

By J. J. Wagoner

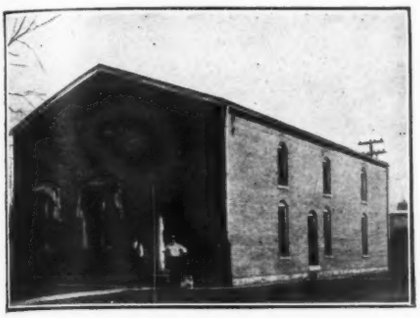
As no continuous, accurate operating record of an American Diesel engine seems to have been kept by anyone for more than one year (the guarantee period), the following account may give useful information not obtainable from any articles published before.

The present equipment of Diesel engines in the Bellefontaine (O.) municipal lighting plant replaced a steam plant consisting of three 100-horsepower shell boilers, two compound engines direct connected to a line shaft from which were belt driven two arc-lighting machines furnishing 91 street arcs and an alternator of 120 kilowatts capacity. The equipment now consists of two Diesel engines of 225 horsepower each, direct connected to two 150-kilowatt General Electric alternators of the revolving-field type, delivering three-phase currents at 2300 volts and 60 cycles. Each alternator drives its exciter by a belt from its shaft; the exciters are Fort Wayne machines of 15 kilowatts and 150 volts.

The street lighting is of the series alternating-current system, using reactive regulators and Adams-Bagnall differential lamps, the arc circuits being supplied from the alternators through ordinary constant-potential transformers. The generators operate in parallel suc-

cessfully, and have not caused any trouble from parallel operation.

As the Diesel engine has been fully described in POWER and other engineering papers from time to time, I do not deem it necessary to go into constructional details and operating principles. The 225-horsepower engines are guaranteed not to use more than 12½ gal-



A SIMPLE TYPE OF CENTRAL STATION

lons of fuel oil per 100 kilowatt-hours delivered at the switchboard, running with a 50 per cent. load factor. The records of this plant show that the economy is much better than the guarantee, as the load we are handling varies from almost nothing to 25 per cent. overload or more. When running at three-quarters load the engines are able to produce power with 9¾ gallons of fuel oil per 100 kilowatt-hours at the switchboard.

Upon starting an engine of 225 horsepower it is necessary to put in the crank case about thirty gallons of oil and thereafter three or four gallons per week for continuous operation. About 70 per cent. of water is kept in the crank case to keep the oil from becoming too thick and to reduce the crank case temperature. It is necessary to renew this oil entirely only once a year if the engines are kept in good condition.

Valve trouble, such as pitting and corrosion, familiar to the user of the producer-gas engine, is not found in the operation of the Diesel engine. When the valves are once properly faced and adjusted they do not have to be removed for grinding on account of carbon, as the carbon deposit in the cylinders (when the fuel is properly admitted) amounts to so little that it can be ignored. If the engine is working properly, the exhaust from a Diesel does not soil a piece of white linen held directly over the exhaust pipe.

PLANT PERFORMANCE

Two engineers only are required to operate and keep the plant in order, one day and one night man, at an expense of $175 per month. There being no boilers to clean, ashes to cart, firing, wheeling, weighing and unloading of coal, etc., no unskilled labor is required.

The plant is furnishing to the city free one hundred and fifty-eight 7½-ampere

OUTPUT AND FUEL CONSUMPTION OF DIESEL ENGINES AT BELLEFONTAINE, OHIO, FOR THE THREE YEARS ENDED MAY 12, 1909.

YEAR 1906.

	Night Service.		Day Circuit Began August 20, 1906.	
	Kilowatt-Hours.	Gallons of Oil.	Kilowatt-Hours.	Gallons of Oil.
May	6,800	646		
June	13,600	1,292		
July	21,652	2,633		
August	31,630	43,629	1,140	566
September	38,380	5,331	3,070	1,312
October	50,010	5,731	7,420	1,607
November	50,660	5,840	10,420	1,802
December	60,510	6,534	14,600	2,060
Totals	274,242	32,336	36,650	7,347

YEAR 1907.

January	55,670	6,259	13,450	2,009
February	47,640	5,398	9,870	1,595
March	50,250	5,720	9,770	1,664
April	45,210	5,230	9,010	1,604
May	40,920	5,009	8,240	1,566
June	38,200	4,628	8,570	1,501
July	40,410	4,608	9,510	1,624
August	46,670	5,418	9,400	1,563
September	49,650	5,777	10,010	1,580
October	63,410	6,841	15,620	2,026
November	65,030	7,107	19,060	2,361
December	76,850	7,982	23,870	2,727
Totals	619,910	69,979	146,380	21,824

YEAR 1908.

January	68,880	7,392	19,550	2,480
February	64,090	6,760	18,060	2,101
March	60,600	6,720	14,430	1,943
April	52,050	6,073	13,450	1,857
May	47,470	5,731	12,015	1,784
June	42,330	5,385	9,895	1,666
July	44,570	5,385	10,675	1,759
August	50,500	5,931	12,170	1,751
September	52,250	6,018	11,520	1,720
October	53,880	6,190	14,920	2,133
November	62,080	7,013	17,010	2,247
December	69,630	7,766	21,740	2,723
Totals	734,330	76,612	175,435	24,166

YEAR 1909.

January	66,080	7,224	19,190	2,395
February	52,980	6,129	13,680	1,863
March	58,710	6,822	15,100	2,071
April	49,280	5,864	12,650	1,818
May	16,870	2,120	4,450	681
Totals	243,840	28,160	65,070	8,850

	Night Service.		Day Service.	
	Total Kilowatt-Hours.	Total Oil.	Total Kilowatt-Hours.	Total Oil.
1906	274,242	32,336	36,650	7,347
1907	619,910	69,979	146,380	21,824
1908	243,840	28,160	65,070	8,830
1909	734,330	76,612	175,433	24,166
Totals for three years	1,872,322	207,087	423,533	62,167

SUMMARY.

	1906.	1907.	1908.	1909.
Kilowatt-hours output	310,892	766,290	909,763	308,910
Gallons of fuel oil	39,683	91,803	100,778	36,990
Gallons per kilowatt-hour	0.13	0.12	0.11	0.12
Gallons of crank-case oil	240	360	295	101
Total cost of fuel oil	$1,190.49	$2,754.09	$3,023.34	$1,109.70
Total cost of crank oil	69.60	87.00	85.55	29.29
Total cost of repairs	23.00	56.00	101.60	27.15
Total cost of attendance	1,345.00	2,100.00	2,100.00	755.00
Total operating costs	$2,628.09	$4,997.09	$5,310.49	$1,921:14
Average load factors	32%	47%	56.6%	45%
Average cost per kilowatt-hour	0.85c.	0.65c.	0.59c.	0.62c.

Average load factor for the entire three years	46 7%
Average operating costs per kilowatt-hour for the three years	0.65c.

street arc lamps and 400 incandescents in parks, alleys and public buildings. It also places $300 per month in the sinking fund, after paying all expenses. This amount is derived from power service at from 2½ to 4 cents per kilowatt-hour and lighting service at 8 cents straight.

Tests of an Exhaust-gas Water Heater

Much has been written and more has been said on the subject of utilizing the waste heat of a gas engine. As we have pointed out several times, the most obvious available method of doing this is to heat (not evaporate) the jacket water discharged from the engine, by means of the exhaust gases of the engine, in exactly the same way that boiler feed water is heated by exhaust steam in a closed heater.

RESULTS OBTAINED IN HEATER TESTS.

	Test 2.	Test 3.
Engine load, brake horsepower	201	100.8
Gas per hour, cubic feet	2,130	1,411
B.t.u. per hour to engine	2,057,600	1,363,000
Pounds water passed through jackets and heater per hour	13,630	5,050
Temperature of water entering jackets	47°	47°
Temperature leaving jackets	100°	140°
Temperature leaving heater	136°	196°
B.t.u. per hour to water in jackets	722,390	469,650
B.t.u. to water in heater	490,680	282,800
Total heat added to water	1,213,070	752,450
Heat represented by work	511,545	256,536
Heat unmeasured	332.985	354,014
Efficiency of engine and heater in utilization of initial heat	84%	74%
Proportion of normally wasted heat conserved by heater and engine jackets together	78½%	68%
Proportion of normally unavailable heat conserved by heater alone	59½%	44½%

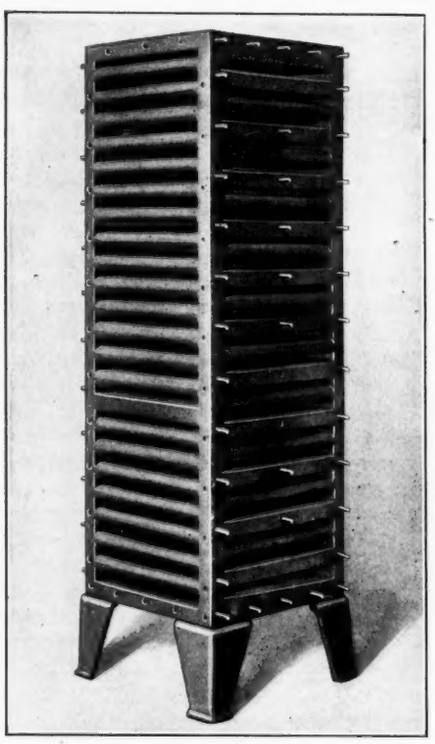


FIG. 1. INTERNAL PART OF EXHAUST-GAS HEATER

The accompanying table gives the principal results of two tests made recently along the line indicated, using a heater built especially for this kind of service by the Williams Tool Company, Erie, Penn. The heater was supplied with the exhaust gases from a Westinghouse 200-horsepower vertical engine using natural gas of 966 heat units per cubic foot at 62 degrees Fahrenheit. The tests here referred to were made at as nearly full load and half load as the load could be adjusted.

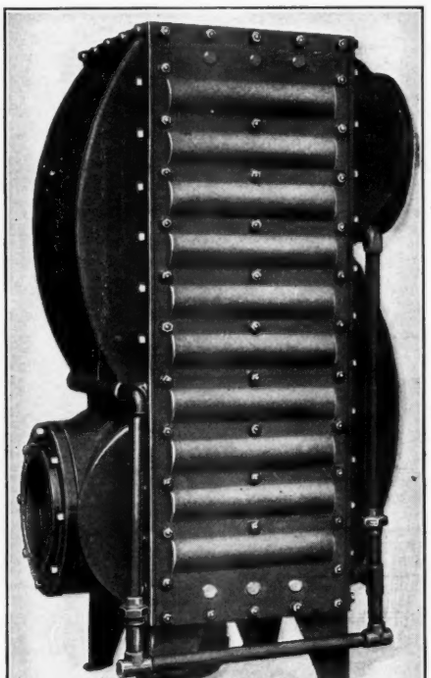


FIG. 2. COMPLETE EXHAUST-GAS HEATER

As the table shows, the efficiency of heat utilization by the engine and heater together was 84 per cent. at full load and 74 per cent. at half load. This is not quite an accurate statement because the heat imparted to the water was probably not all utilized or even available. But it was at least conserved and made largely available for heating purposes.

The proportion of the otherwise useless heat conserved by the heater was 59½ per cent. at full load and 44½ per cent. at half load. That is to say, of the heat represented by exhaust-gas temperature, engine friction, radiation, etc., but not including the jacket water, the heater made available 59½ and 44½ per cent. at the two loads stated. Unfortunately no figures are at hand which show the individual efficiency of the heater itself.

The heater consists, broadly, of a series of horizontal partitions cast between four uprights forming a square frame, as illustrated in Fig. 1, so as to form two sets of horizontal passages between them; the passages of one set running at right angles to those of the other set and alternating with them from the top to the bottom of the heater. The transverse water passages, constituting one set, are connected together by pockets in the outer casing and the gas passages are connected by larger curved flues in the casing. Reference to Fig. 2 will give an idea of this feature of construction. The heater with which the tests were made contained 134 square feet of heating surface, which is about right for service in conjunction with a 150-horsepower engine; in other words, it was about one-third smaller than it should have been to obtain the best economy. This was made manifest by the fact that the gases escaped from the heater during the full-load run at an average temperature of 238 degrees, Fahrenheit. Using the countercurrent principle, it would not be necessary to have so much difference between the escaping gases and the entering water.

Gas Power Investigation for the Navy?

In his annual report, H. I. Cone, chief engineer to the Bureau of Steam Engineering, United States navy, makes the following statements and recommendation:

"The development of large gas-engine machinery operated in connection with gas producers is rapidly progressing. The bureau is of the opinion that this prime mover is worthy of much more serious investigation and experiment than have yet been given it in our navy.

"It is possible to design a marine plant composed of bituminous coal-gas producers furnishing power to gas engines. There are several such plants already in existence; although they are small, they prove the feasibility of extending this system of developing power to larger installations. Great possibilities for improving the economical operation of machinery are offered by the producer-gas engine combination. We cannot afford to delay development of the gas engine for naval use until commercial gas-engine plants, directly adaptable to naval use, are regularly on the market, and it is, therefore, recommended that special authority be secured in the next appropriation act for the diversion of not more than $250,000 of the appropriation "Steam Machinery" for the purchase and installation of a producer-gas engine plant in one of our colliers, if at any time during the year such an expenditure is deemed advisable."

The Illinois Steel Company is putting in an 8000-kilowatt mixed-flow turbine, guaranteed to run on 25 pounds of steam per kilowatt-hour, with an absolute initial pressure of 16 pounds and 28½ inches of vacuum.

FROM READERS *with* SOMETHING TO SAY

PRACTICAL INFORMATION *from the* MAN *on the* JOB: A LETTER GOOD ENOUGH *to* PRINT *is* GOOD ENOUGH *to* PAY *for*

Water in the Cylinder

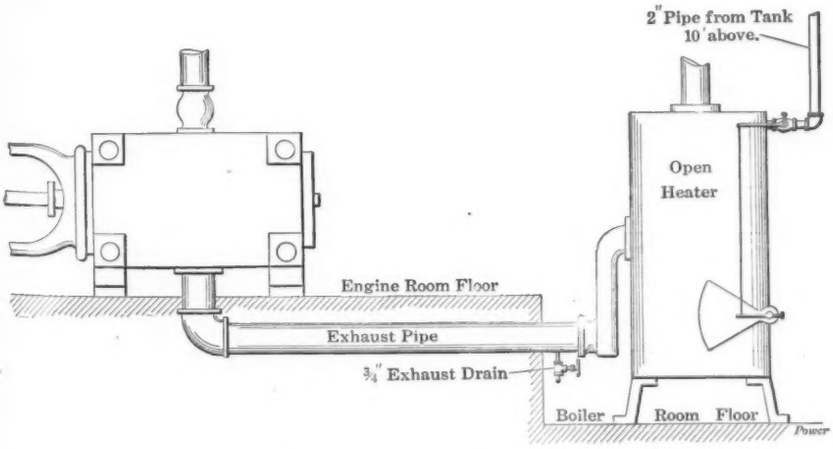
Some years ago I erected a 22x48-inch Corliss engine in a cotton mill in Louisiana. Like all such engines, it carried a large flywheel with a heavy rim, grooved for 2-inch ropes. As the mill machinery was not yet ready, I tested the engine by starting up light, with only the line shafting. The buyers were satisfied, gave me an acceptance and I went to other fields.

About three or four months later, I received information that the engine had been wrecked, and was directed to proceed at once to the spot and ascertain the cause. The mill had not yet begun operating, and the engine on the day of the wreck was being used for a few moments only to turn the shafting while belts were being run on. The engineer closed the throttle in the usual way, and after speed had diminished, uncoupled at the wristplate, inserted his retarding bar, and in the usual manner reversed the valve gear to bring the engine to a stop off the center. It was at this juncture that the wreck occurred.

I immediately realized that nothing but water in the cylinder could have broken the piston, stripped the rod from the crosshead, bent the connecting rod into an S-shape and loosened the engine from its foundation. But I was unable to figure where the water came from. I dismissed the boiler-foaming theory at once, as the accident occurred after the throttle had been closed and not enough steam was being taken to draw any quantity of water over into the cylinder. Obviously, the water must have come from the other direction. Like a flash, I remembered an incident told to me by an old engineer, about an engine acting as a pump after steam had been shut off and drawing water up through the drain pipes running into a sewer.

I had no sooner arrived at the mill than I made straight for the heater in the boiler room, only pausing a moment to glance at the engine, a move that somewhat surprised the onlookers, who had been waiting for my arrival.

The accompanying sketch explains clearly the cause of the accident. I looked the heater over thoroughly and found no drain. There was, however, a ¾-inch drain on the exhaust pipe. I asked the fireman if this drain had been open at the time of the accident, and, as expected, received the answer "yes." My quick move to the boiler room had put everyone on their guard. As the drain opened into the room, steam would have escaped in uncomfortable volumes and I concluded that the chances were the valve had been closed. However, it would have made no difference in the result had it been open, for with the automatic float valve stuck open, as it undoubtedly was at the time of the accident, the 2-inch pipe under 10-foot head would have admitted water much faster than the ¾-inch pipe could have let it out.



ARRANGEMENT OF PIPING

To further strengthen my theory, I learned that the engine had not been run more than a very few minutes, and then not up to speed, so it was not unreasonable to suppose that the exhaust pipe was full of water.

W. D. ALLEN.

Yazoo City, Miss.

Packing Water Pistons of Small Pumps

When packing the water end of a 2-, 2½- or 3-inch boiler-feed pump, have you ever wished that your hands were smaller or that the pump cylinder was larger?

You will notice I have given cylinder sizes which correspond with pipe sizes. In most pumps you will find that the counterbore is slightly smaller in size than the outside of a piece of pipe of the same nominal size. If a short piece of the same-sized pipe is turned or filed down to fit the counterbore you will have a continuation of the bore of the cylinder.

Cut and fit the packing to the pipe and put the pipe into the counterbore; then with a longer piece of pipe of the next smaller size you can push the ring of packing from the pipe to its place on the piston without having the joint come apart.

W. C. REED.

Pittsfield, Mass.

Testing Boilers with Hot Water Meter

The writer has seen several boiler tests conducted by theoretical engineers, and each case was conducted according to the views of the engineer in charge. Most engineers have an impression that if the feed water is not weighed by the generally adopted method, the accuracy of the test will be impaired. This method is to place three barrels at the water end of the feed pump, two of them being elevated above the third. The city water is admitted through valves into the two upper barrels, separately, and in turn led through valves at the bottom into the lower barrel, from which the suction from the pump is taken. Before beginning a test the weight of water in each of the top barrels is ascertained, and by noting the number of times they are filled, the total weight pumped can be easily found.

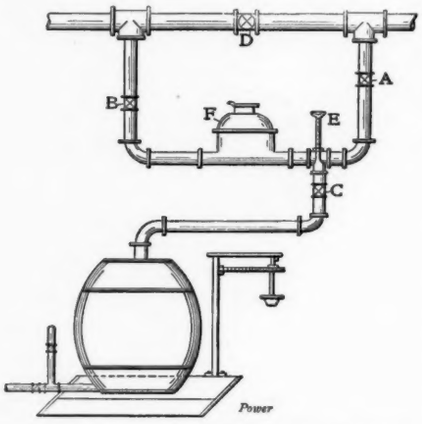
In the writer's opinion, this method is good, but should be used only on short tests; on long tests it necessitates hard labor and extra expense, besides the liability of making a mistake while the water is changed from one barrel to the other. The cost of installing and connecting the barrels with the valves and piping will exceed that of a hot-water meter.

There are in the market several machines for weighing the feed water, and

the manufacturers guarantee that they will work within 2 per cent. of the accurate weight. I do not deny that these machines will work all right when they are new, but after being used for some time they become worn and are liable to leak. These machines are three times as expensive as a hot-water meter, and occupy more space.

The writer is conducting some tests in which he is using hot-water meters, and is obtaining results just as accurate as with any scale on the market. The meter can be connected either in the main feed-water line, or in the branch feed pipes of any boiler on a battery which it may be desired to test.

Fig. 1 shows how the meter may be checked to determine its accuracy. The



CONNECTIONS FOR CALIBRATING METER

valves *A* and *D* are closed and *B* and *C* are opened, until the barrel is full. Before the water is allowed to pass to the barrel, the readings of the meter *F*, and the thermometer *E*, are taken. When the barrel is full, the valve *C*, is closed and the barrel weighed. The operation should be repeated at least ten times, and the first and last readings of the meter carefully noted.

Suppose that the first reading of the meter was 425 and the final reading 475, then it would appear that 50 cubic feet of water had passed through it. Let the average reading of the thermometer be 180 degrees Fahrenheit. The weight of 1 cubic foot of water at that temperature is 60.6 pounds, which would make the total weight as indicated by the meter, 3030 pounds. Assuming that the 10 barrels of water weighed 3185 pounds, the meter would read low by 155 pounds, and the percentage of error would be

$$155 \div 3185 = 0.05 = 5 \text{ per cent.}$$

By knowing that the meter leaks 5 per cent., the true reading can be found by dividing the indicated amount by 0.95.

Suppose that the first reading of the meter was 4125, and the final reading 4195; then it would indicate that 70 cubic feet of water had passed through. Let the average reading of the thermometer be 160 degrees Fahrenheit, the weight of 1 cubic foot of water at that temperature being 61 pounds. By multiplying the number of cubic feet passed through the meter, by the weight per cubic foot, we obtain the total weight in pounds, thus

$$70 \times 61 = 4270.$$

Assuming that the water which was weighed in the barrels was 4150 pounds, then it shows that the meter does not leak, but is registering 120 pounds more than it ought to. In this case the error is

$$120 \div 4150 = 0.029$$

or 2.9 per cent.

To find the true reading multiply 4270 by 0.971.

VASIL MACKAY.

Boston, Mass.

The Gyroscope

Happening to open one of the early numbers of POWER, that of November, 1905, the first words that I saw were "The Gyroscope," as the title of an article, and I could not do otherwise than read, carefully, the whole of it.

I had spent much time in mental and physical work upon this curious device. The first thing I learned was that the axle of the wheel, while revolving at a high speed, would carry on the projecting end a surprising amount of weight without losing its balance. For this action there was no readily seen reason. When the wheel slowed down, the weight descended; when made to speed up, the weight rose again, close to the vertical position, and the supporting frame would move slowly around.

Quoting from the article: "This angular deflection is best called 'rotation,' to distinguish it from movements of 'translation.'" But I do not see how anything can be said to "rotate," when the movement is partially angular only. It looks to me like "translation" in a line of curvature similar to an automobile turning a corner.

In artillery and small-arms practice, rotation and translation work together to deflect both the lateral and vertical motion of a projectile, and to obtain a quicker motion. Rifling will change the lateral position of any projectile, as is well known among experts in rifled arms.

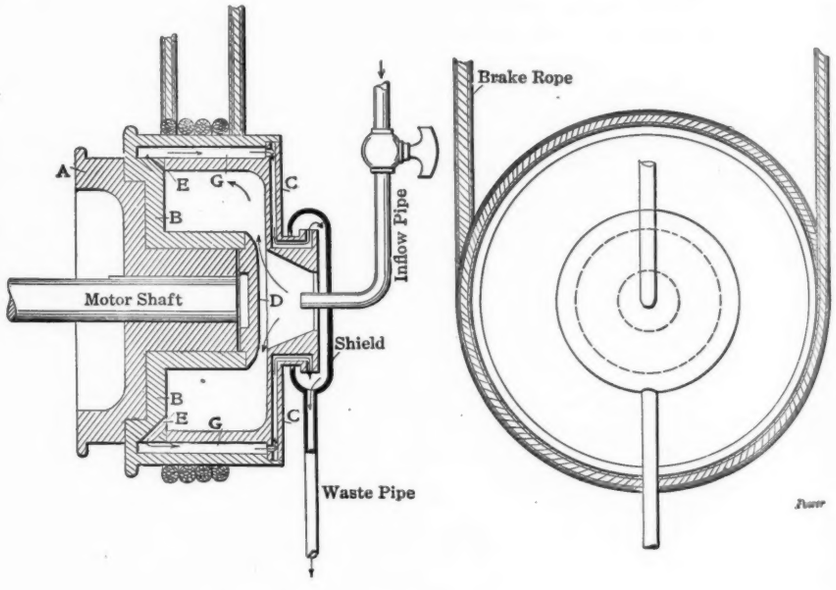
For many years the gyroscope gave me much study and thought. The boys' peg top gave me the first idea as to why it remained in an upright position so long. The next helper was the rifled gun. Thereupon came various-sized artillery projectiles, all piercing the air spirally. When they spun to the right, they curved horizontally in that direction; when the spiral was reversed, so was the direction of the projectile.

PETER VAN BROCK.

Jefferson, Ia.

A Method of Cooling Brake Pulleys

The accompanying illustration shows a form of water-cooled brake pulley which was successfully used in making tests on a number of motors of small capacity. A rope brake was used in making these tests, but the scheme can be used equally as well with other forms of absorption brake. The pulley used was 7 inches in diameter, but there is apparently no reason why the method cannot be successfully used on larger pulleys, in testing machines of greater capacity.



SEVEN-INCH WATER-COOLED BRAKE PULLEY

It will be noticed that the pulley is made of several parts, and is so fitted together that first-class workmanship is required to get satisfactory results.

The part *A*, which also serves as a pulley for the speedometer belt, is keyed

to the shaft of the machine, and on the hub of this is mounted the brass piece *B*, which comprises the face of the pulley. The end of *B* is turned down on the inside to receive the cap piece *C*, which extends back and butts firmly against the shoulder on the inside of *B*. A small brass piece *D* is placed on the end of the hub to protect the shaft from the water.

It is evident that water admitted through the inflow pipe will follow the course indicated by the arrows, due to the fact that when the pulley is running, the centrifugal action will force the water through the holes marked *E*, and into the annular space between the rim of the pulley and *C*, and out through the holes into the shield. The fact that the outlet holes in *C* are not far removed from the center will permit the pulley to run at a high speed without an undesirably high centrifugal force.

There may be four or more of the 45-degree holes leading into the annular space *C*, and a like number of discharge holes leading out through *C*. The number of holes, of course, will depend upon the speed of the pulley, as a high speed will secure ample circulation with fewer holes than a slower speed. In the tests made four holes were used, and by varying the supply of water, a proper circulation could be secured.

It will be seen from this that a constant, easily regulated circulation can be obtained through the annular space on the inside of the rim, and that the water is admitted and carried off in such a way as to prevent the annoyance of having water splashing over things.

L. S. Le Tellier.

Charleston, S. C.

Why Do Not Engineers Use the Slide Rule?

It would be a great advantage to practical engineers if they would ask themselves the above question.

Most of us will say, "Yes, I know it is a fine thing, but it takes a man well up in mathematics to be able to use one." Why do we say this? It is because we see only a few men using them and these we know to be educated. I happen to know that anyone can operate the slide rule for the simple reason that I can do it myself, and I do not know half as much about books as the average engineer. I learned to use one by carrying it with me and every time I had a few minutes to spare, would try to figure it out.

One day a stranger came along and showed me how to multiply 2 by 2. He said that that was all I had to learn about it and he proved to be right, for from that time on I have had no more trouble.

There is no mystery about it at all. It is not necessary to know anything about logarithms and, in fact, I do not believe that one slide-rule operator in a thousand does. One trouble is that the manufacturer issues a book of instructions which we engineers cannot understand, one of its first statements being that the slide rule is based upon a logarithmic scale and must be understood as such. This usually discourages the average man and he gets no further.

To make it still harder, perhaps he has purchased a compound rule. This the beginner should never do, although they are easy after one has learned on the simple kind, which can be bought for a small sum.

Engineers should find the study of the slide rule interesting, and after they have become familiar with it, should be able to use it for all kinds of calculations such as are met with in everyday work. The writer has taught several engineers how to use it, in less than two hours, and they all seem to be surprised to find it so easy.

If any man who has a rule cannot use it, I should like to hear from him and will be glad to assist him.

H. T. Fryant.

Jackson, Miss.

Automatic Compensating Bearing

In the maintenance and care of vertical turbine water wheels much annoyance is encountered by the tendency of the step to wear down. This usually does not give much trouble on low heads, because wheels are so made that the pressure of the water coming from the under side helps to support them. But where the head of water is high, and a long and heavy length of shafting rests on the wheel, the help received from this source is not sufficient to keep the wheel up. Notwithstanding the fact that in such cases collars are usually put onto the shaft to help keep it in position, the wear is continually downward; and as most wheel pits are wet, the water dripping and splashing on the collars makes lubrication with oil impracticable. This necessitates a constant stream of water running on the collars, to keep them cool. This is done by means of a piece of pipe screwed into the penstock opposite each collar and in such a position as to direct the water onto the collar. When the water is turbid, the grit has the effect of a grindstone, resulting in the rapid wear of the collar.

To overcome this trouble I arranged a device called a compensating bearing, an assembly of which is shown in the lower part of the figure. It has been in successful operation for several years, holding up a water wheel having 75 feet of 3-inch shafting, which with pulley and couplings makes a weight of more than a ton. It is located above the wheel pit and near the top of the shaft; thus it is removed from the wet and can be lubricated with oil.

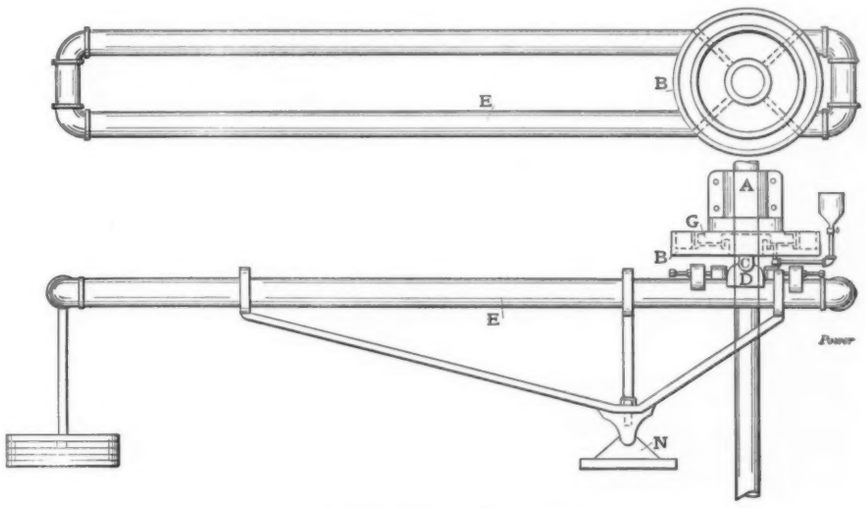
Referring to the figure, *A* is a split collar rigidly fastened to the shaft; the lower part of this collar is in the form of a bearing flange *G*. The loose collar *B* is made with two separate compartments and serves to hold the cooling water. The inner one is the bearing for the split collar and is provided with oil grooves as shown. The collar *B* is provided with two lugs *C*, one on either side, which rest in semi-circular bearings *D* fitted loosely on the frame *E*, which is composed of 2½-inch pipe. The frame is trussed as shown and rests on the pivot bearing *N* which is supported on the concrete foundation of the turbine. At the outer end of the frame *E* a suitable number of weights are attached to counterbalance downward pull of shaft.

Upon starting the turbine, when the gate is first opened, the frame oscillates considerably but gradually settles down to a state of equilibrium.

This device has been in use for five years and has needed no attention other than to keep it properly lubricated.

W. H. Neal.

Philmont, N. Y.

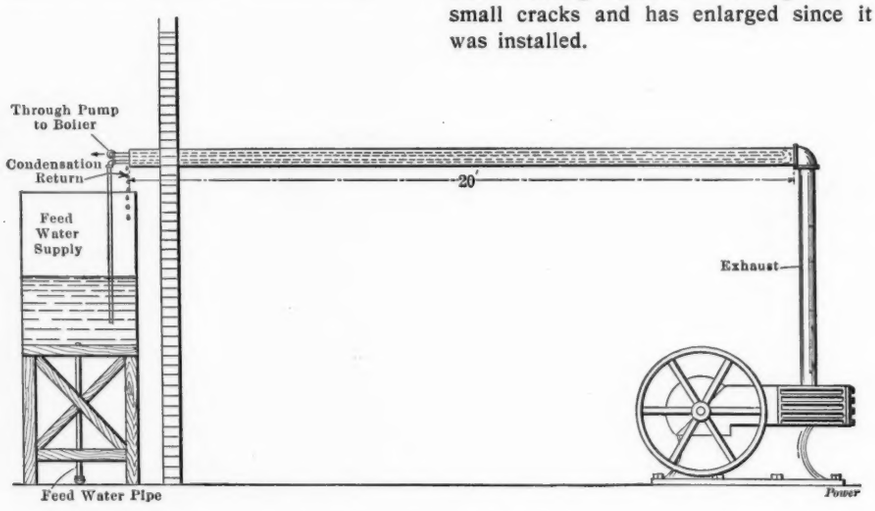


Compensating Bearing

A Novel Feed-water Heater

While passing through a small lighting plant, I noticed a type of feed-water heater which may appeal to some engineers of small plants who cannot persuade the owners to invest in more expensive heaters.

The generating unit is a 12x12-inch Ideal engine, belted to a General Electric alternator. The engine exhausts through a 4-inch pipe to the atmosphere. The exhaust line is carried across the engine room to the exterior wall of the building. The feed-water tank is directly under the end of the exhaust, and is supplied by a duplex pump, with water from a distant stream. As shown in the illustration, the 1-inch feed-water pipe enters the end of the exhaust, passing back to the ell, a distance of 20 feet, and out again by means of the return bend.



ARRANGEMENT OF PIPING

It is claimed that feed water at 50 degrees Fahrenheit in the tank is delivered to the boilers at 205 degrees Fahrenheit, which, according to "Kent," excels the performance of the average heater.

To find the B.t.u. transmitted from the exhaust steam to the feed water in one hour, we have the following data:

Horsepower of engine, 80.

Average load in horsepower, 60.

Water consumption (assumed), 25 pounds per one horsepower-hour.

Area of heating surface, 10.5 square feet.

Rise in temperature of feed water, 155 degrees Fahrenheit.

The total water consumption per hour will be

$$60 \times 25 = 1500 \text{ } \textit{pounds.}$$

The B.t.u. given to the 1500 pounds of feed water in an hour will, therefore, be

$$1500 \times 155 = 232{,}500$$

B.t.u. total, or 22,150 B.t.u. per square foot of heating surface.

Following an approximate rule that "a saving of 1 per cent. is made by each increase of 11 degrees in the feed water," we find a saving in this case of 13.7 per cent., but radiation loss would cut this down to about 6 or 7 per cent.

R. S. SEESE.

Urbana, Ill.

Fittings for Superheated Steam

Power-plant operators have trouble with the valves and fittings they have to use, and as the temperature rises the more trouble is experienced. Many fittings and valves will fail under 250 degrees of superheat. While it has been recommended by some engineers to use an iron of 30,000 pounds tensile strength, this does not always insure our getting a good, strong or suitable casting for high-pressure work. Growth takes place in fittings that are not made of the right composition and we may find that we have a fitting which is becoming full of small cracks and has enlarged since it was installed.

Most any fitting will give good service under a pressure of 110 pounds, but as we raise the pressure and use superheat we encounter trouble. Some are recommending the use of steel castings, but a good cast-iron fitting will answer the purpose and will do quite as well when made of the proper mixture.

Cast iron contains silicon, sulphur, phosphorus, manganese and carbon. By varying the proportions of these we can make an iron suitable for grate bars, steam fittings or flywheels. From many careful tests it has been found that it is necessary to have low silicon, low phosphorus and low carbon in iron for fittings for superheat work. The following specification is about right: Silicon, 1.72; sulphur, 0.085; phosphorus, 0.89; manganese, 0.48; total carbon, 2.50; combined carbon, 0.17 per cent. A fitting made from this iron will stand 300 degrees superheat. One of the best mixtures is the following: Silicon, 1.40; phosphorus, 0.23; sulphur, 0.06 to 0.08; manganese, 0.50 to 0.75; total carbon, 3 per cent.

C. R. MCGAHEY.

Sheffield, Ala.

The Calorimeter

It is generally considered that the steam calorimeter is almost a useless device, unless it is carefully used. But within the past year the steam tables have been revised and have given a precision within one-twentieth of 1 per cent. Why then can we not now rely on our calorimeter determinations?

Priming in the boiler depends on the kind of boiler, the kind of water used and the rate of generating steam. Even when steam is given off at a very slow rate there are very minute particles of water carried up with the steam and into the steam mains. We probably shall never be able to generate dry steam in an ordinary boiler without a superheating arrangement. The amount of this moisture is variable, and may be anything from a small fraction of 1 per cent. to 5 or 10 per cent. In a boiler test, however, the main object is to get a sample to examine. If the pipe from which we take our sample is horizontal (there must be water of condensation with the very best of pipe coverings), the condensation will run down the lower part and sides of the pipe. If we can tap our pipe line at a point where the steam is going vertically upward and take the sample from the middle, the chances are very favorable that we can get a good fair sample of steam. The throttling calorimeter is a device that will give very fair results, and we now have steam tables that give us greater precision than we can get in our observations in the boiler room. If care is taken in procuring the sample sufficient accuracy can be obtained for most purposes.

H. J. MACINTIRE.

Cambridge, Mass.

Trouble with Water Supply System

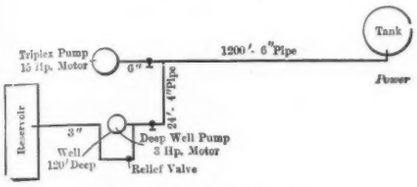
The accompanying diagram shows a layout of water-supply piping for an institution in which I am interested.

The water is pumped by a 7x8-inch Goulds triplex pump operated by a 15-horsepower motor, and is delivered to a tank 75 feet above the ground, through a 6-inch main, 1200 feet from the pump house. Underground the 6-inch pipe has a fall of 25 feet to the pump, the gage showing a pressure of 42 pounds at the pump. This arrangement works nicely and is in daily operation until 9 p.m., at which time it is shut down until morning.

Recently complaints have been made that there is very little water in the tank in the morning. To overcome this difficulty a deep-well pump was put into commission and operated all night to keep up the supply. The pump is 3 inches in diameter by 16 inches stroke and 35 revolutions per minute, and is driven by a three-horsepower motor. The well is supposed to be 120 feet deep. The delivery pipe from this is 4 inches and connects into the 6-inch main from the

triplex pump. There is also a 3-inch branch to the reservoir. A relief valve set at 50 pounds is placed on this branch, to operate in case of the tank becoming full and the ball-cock closed.

From my own observations, when the 4-inch valve from deep-well pump is closed, the relief valve opens and allows



LAYOUT OF PIPING

the water to discharge into reservoir. When the 4-inch valve is open the pump works easily and does not discharge into the reservoir.

Will this pump deliver water to the elevated tank and in what quantity? There are differences of opinion on this question. The chief engineer claims that it will not work, while the assistant engineer states that he has seen water flowing into the tank. I was not in a position to reach the top of the tank myself, and would like expert opinion on this subject.

D. M. MEDCALF.

Toronto, Ont.

A Choked-up Drip

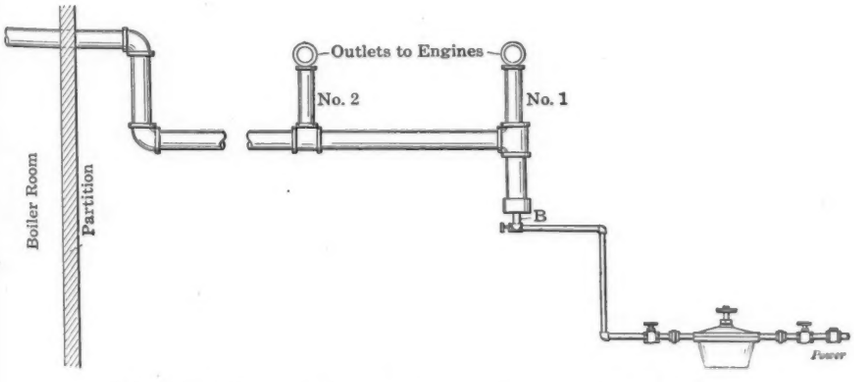
Whenever an engine gets water into the cylinder, after opening the cylinder drips, there is always a mad rush to the boiler room to find out if any of the boilers are priming, or if the fireman has been neglectful and let the boiler get too full, thereby causing the water to pass over with the steam in dangerous quantities. The fireman is not always to blame, however; sometimes the trouble lies in the engine room, as was the case in the following instance:

The plant was equipped with two engines of a high-speed type, and it was plainly noticeable whenever water started to enter the cylinders. No. 2 engine was not noticed so much, but it became impossible to run No. 1. The bypass on top of the trap was opened, and also the bypass in the drip line, but it did not alter conditions. As everything was normal in the boiler room, it was necessary to look elsewhere for the cause. The trap was disconnected and the valve in the drip line under the header was opened, but not a sign of steam or water was apparent. The main steam valve was then closed and the drip pipe *B*, shown in the illustration, was taken apart; there the real cause of the trouble was found. The section of pipe below the tee was partly filled with scale, broken valve disks, etc.; the drip pipe and valve were entirely choked and prevented any of the condensation to pass away through the trap.

We are eternally looking for traps that allow steam to pass through, but how about the ones that cannot perform their duties by the pipes leading to them being stopped up?

EBENEZER WHITAKER.

New York City.



PIPING, SHOWING DRIP THAT WAS CHOKED WITH SCALE

Engine Troubles

How many engineers have ever seen an engine operate with one cylinder head off? I recall the case of a 50-horsepower, high-speed, automatic-cutoff engine which knocked persistently in spite of everything that was done. The rear head was removed and the engine started under load. When starting, steam poured from the exhaust ports in great volumes, but after coming up to speed the escaping steam was not so plentiful, and it was possible to stand quite close to the open head and observe the piston at every stroke. At full stroke the steam did not roar or hiss, but in escaping, made a noise exactly like the crack of a pistol; the knocking was much exaggerated. It was found that the piston had worn several hundredths of an inch smaller than the cylinder, and that it lapped the steam port half an inch. The sudden rush of steam caused it to strike against the opposite side of the cylinder and thus make the knocking sound. In proof of this the cylinder was rebored, a new piston was installed, and good rings put in, with the result that the knock was never again heard.

In another engine in the same plant a knock was observed which occurred only at every few revolutions and at irregular intervals. It was a sharp knock at the in-stroke, followed by a dull thump on the out-stroke. It was eventually ascertained to be a loose piston, which would strike first the nut and then stick on the taper for a few revolutions and then fly back again.

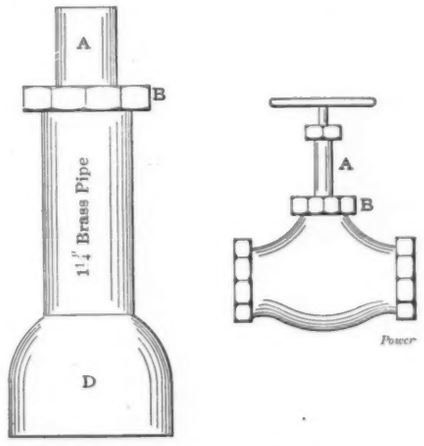
I have used an ordinary physician's stethoscope with excellent results in studying the sounds of an engine. A piece of ½-inch pipe with a tube attached to it is almost as good. By placing the pipe against the steam chest or other place where a sound is to be studied, and the tube in the ear, it can be heard more plainly. Knocks and other sounds that seem to travel all over the engine can often be located by this method.

E. N. PERCY.

San Francisco, Cal.

Homemade Torch

A torch is a very useful thing in a steam plant not equipped with electric lights, especially where there are several gas jets to be lighted. It appears that the only torch in our plant belonged to me and was kept in the engine room.



THE HOMEMADE TORCH

Oftentimes, especially Monday mornings, my little lighter would be missing and this meant that I had to go the rounds and light about 15 gas jets with matches. I concluded to make a torch of my own and keep it locked up so as to be sure I would have it for use in case of an emergency. The figure illustrates its construction. The bottom *D* was made from an old grease cup. I cut off the lower part of the cup and drilled a hole in it to take a 1¼-inch light-weight brass pipe. The nut *B* that screws into the pipe and the sleeve *A* were taken from the old valve shown to the right of the figure.

W. A. DOW.

Cambridge, Mass.

Removing Piston Rod from Crosshead

It frequently happens that the piston rod of an engine must be removed from the crosshead, and where the rod is fastened by keys it is sometimes quite a task. Rods of this kind usually have a tapered seat, in assembling, and they are

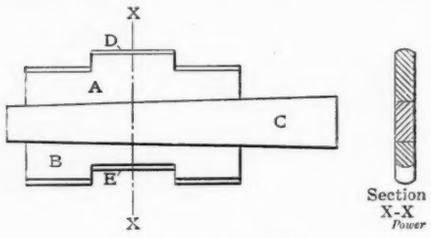


FIG. 1. PULLING KEY

sometimes drawn into the neck of the crosshead so tightly that considerable difficulty is experienced in drawing them out again.

In our station two methods are used: the familiar pulling key and the jack, both being shown in the accompanying sketches. The use of these keys is familiar to most engineers, but for the benefit of those who have never used them a brief description may be of interest. Key *A* is placed in the key slot with its projection *D* against the end of the slot in the rod nearest the piston. Key *B* is placed in the slot with its depression *E* opposite the other end of the slot. Then by driving the wedge-shaped key *C* between *A* and *B*, the rod is

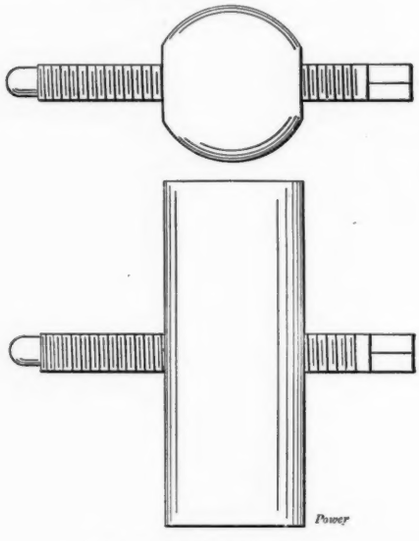


FIG. 2. PISTON-ROD JACK

forced out of the crosshead by the projection *D*.

Where the rod will not respond to the efforts of the pulling keys the jack may be used. To do this it is necessary to remove the crosshead pin and shove the crosshead with the rod and piston back from the connecting rod. The heavy piece of the jack is then put in place of the crosshead pin and the screw inserted by putting it in from the rear through the connecting-rod opening in the crosshead. The screw is run up until its point strikes the end of the rod, when by the aid of a wrench on the head of the screw the rod is forced out. The greater the number of threads on the screw the more powerful the screw becomes. Where the side opening between the crosshead guides is not wide enough to permit the use of a wrench on a square-headed screw, the head may be left round and a pipe wrench used.

C. L. GREER.

Handley, Tex.

Engine Troubles

I was called upon to key up a small double-cylinder vertical engine, and found that the key would not stay in place more than a day at a time. Finally an overhauling was decided upon, and one side was disconnected and the other side run as a single engine. The removal of the top head disclosed a mark which had been made by the end of the piston rod striking the cylinder head. The bottom end of the cylinder had a shoulder worn about ½ inch wide, on account of the piston never having traveled over the counterbore.

The cylinder was rebored and as much as was deemed safe, was cut off both ends of the rod, reducing the brasses to make up the required amount and fitting liners to the opposite sides.

Again, on looking over a small automatic engine, I was startled to hear it begin to pound violently just as soon as the steam was shut off. Stopping it and starting again resulted in the same pound, until the governor took hold and the throttle was opened wide, when no pounding was heard. On shutting off the steam again, the pound was in evidence until the engine stopped. Placing the engine on its back center, we disconnected the rod at the crank to test the clearance, and, as expected, found the piston hard against the back head. A proper change of liners equalized the travel and the trouble was remedied.

Probably the excessive compression with this class of engine kept the piston from striking when running with throttle wide open.

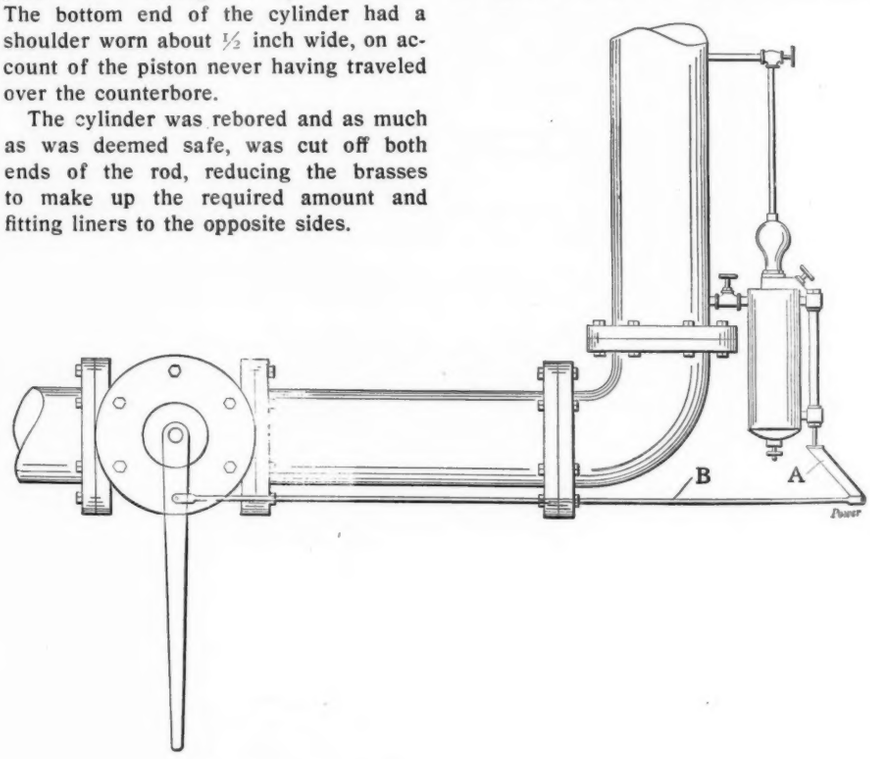
J. V. N. CHENEY.

South Portland, Me.

An Automatic Lubricator

In an effort to reduce the cylinder-oil consumption on a pair of 16x36-inch hoisting engines I conceived the idea shown in the accompanying illustration. It consists of a lever *A* attached to the regulating valve on the lubricator, and a connecting rod *B* leading to the throttle of the engine, thus making the regulation entirely automatic.

The engines hoist an average of fifty cars per hour, requiring twelve seconds per car. This means that the engines are in actual operation only ten minutes out of every hour. Previous to the change, the engineer regulated the feed



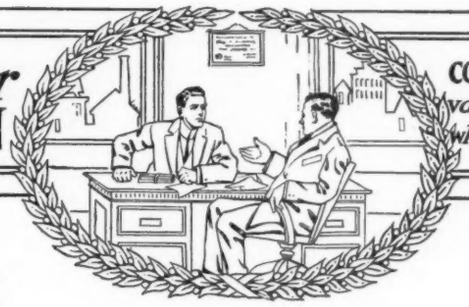
LUBRICATOR WITH AUTOMATIC ATTACHMENT

of his lubricator when he started in the morning and allowed it to feed until he stopped in the evening, thereby consuming one quart of oil for the day's run. Since installing the automatic device, a quart of oil will last one week, a saving of 83⅓ per cent. in the oil bill, in exchange for an expenditure of a little time and thought.

CHARLES J. HALLORAN.

Avoca, Penn.

SUBJECTS *under* DISCUSSION



COMMENT, CRITICISM, *and* DEBATE *upon various* ARTICLES, LETTERS *and* EDITORIALS *which have* APPEARED *in* PREVIOUS ISSUES

"Freak" or "Fake" Diagrams?

In a recent issue, a "freak" diagram was shown, and an explanation of the cause was asked for.

I am not going to attempt a guess as to the cause, for I believe that the man who took that diagram is the one man in the whole world who is in the most favorable circumstances to find the cause of the freak, providing that he does not already know the cause.

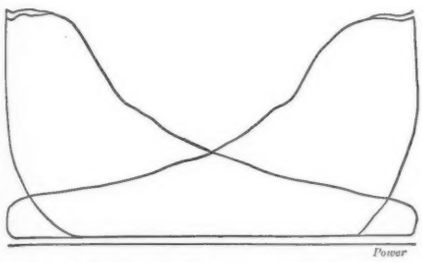


FIG. 1. ACTUAL RUNNING CONDITION

Herewith are shown three diagrams taken from a Buckeye engine. Fig. 1 shows the actual running condition of the engine. Figs. 2 and 3 are "freak" diagrams. To state it a little more correctly, they are "fake" diagrams, because there was absolutely no change whatever made to the engine in any way; the indicator and its connection to the cylinder and the crosshead were not altered in the slightest, yet these diagrams were taken from the same engine as diagrams in Fig. 1, within five minutes after it was taken.

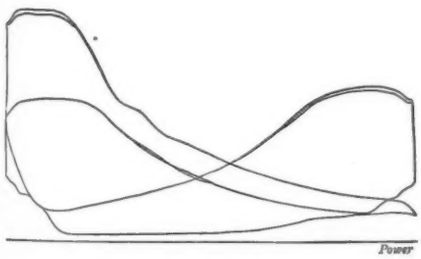


FIG. 2. FAKE DIAGRAM

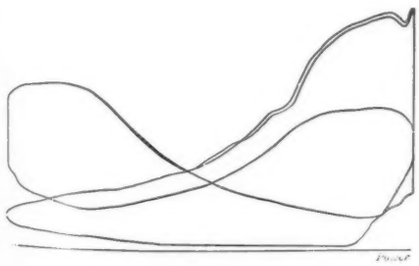


FIG. 3. ANOTHER FAKE

If any of the readers have ever taken any "fake" diagrams, they will know at a glance how these were obtained. But if they have not, they might guess until the bench breaks without hitting upon the true cause of the "freak."

It is quite easy to submit "fake" diagrams if a person wants to make the indicator tell lies.

HARRY W. BENTON.

Cleveland, O.

In explanation of the cause of Mr. Phelps' "Freak Diagram," in the February 15 issue, I would say that the defect is caused principally by pencil friction, and by bringing the pencil in contact with the card during the period of exhaust instead of during the admission period.

By applying the pencil to the card during the period of exhaust, part of the exhaust line, the compression, the admission, the steam, the expansion, the release and finally the unfinished portion of the exhaust line are taken. The last portion of the exhaust line will stand a greater distance above the line of absolute vacuum than the first portion, as shown by diagram in Fig. 1.

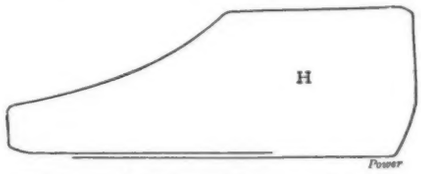


FIG. 1. VARIATION DUE TO PENCIL FRICTION

The distance between the lines is a measure of the defect due to pencil friction and is increased or diminished in direct proportion to the pressure on the pencil point.

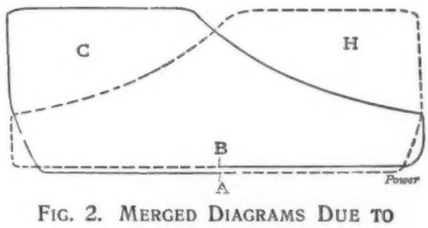


FIG. 2. MERGED DIAGRAMS DUE TO PENCIL FRICTION

In taking the diagram of the opposite stroke, and by first applying the pencil during the exhaust period, the pencil will touch at a point at a distance above the line of absolute vacuum equal to that of the first portion of the exhaust line of the first diagram taken, and merge into it as shown at *A*, Fig. 2, and finally blend into the portion of the first diagram which was drawn last, as shown at *B*. The dotted lines represent the head-end diagram while the full lines represent that of the crank end. Diagrams in Figs. 3 and 4 were taken by the same indicator, and from the same engine, a few minutes after those in Figs. 1 and 2.

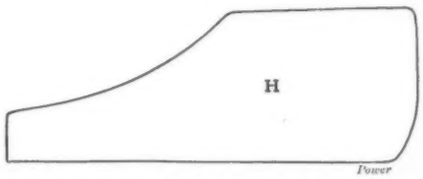


FIG. 3. PERFECT DIAGRAM

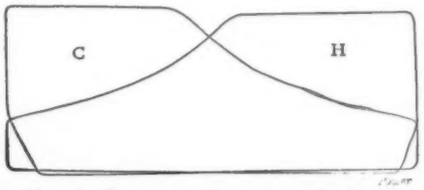


FIG. 4. PERFECT COMBINED DIAGRAM

CHARLES J. HALLORAN.

Avoca, Penn.

In the February 15 number C. N. Philbrick submits diagrams which he wishes to have criticized. About the only thing which appears to be unsatisfactory is the steam line of the head-end diagram, which is not as horizontal as it would be in the ideal diagram. This falling off could be caused by a throttling of the steam, that is, either the steam pipe or the port might be too small for the required duty. A very high piston speed would also have the same effect. However, this last cause would hardly apply to this case, for the piston speed is not excessive. Neither is it likely that the steam port is too small, for there seems to be enough steam supplied to the other end and the ports are, no doubt, of the same size. It would seem, then, that the defect lies either with the action of the valve or with the working of the indicator. If the valve action is slow, it might cause a sort of wire drawing of the steam, and hence a slope in the steam line. On the other hand, the irregularity might be

caused by the indicator pencil being disturbed, or the indicator itself may be defective and unsteady and so give an irregular line.

The thing to do is to take several diagrams, being careful that the indicator is in good shape, and if these are like the first diagram taken, it is pretty safe to say that the trouble is with the valve action. Then the valve should be adjusted and more diagrams taken. After a few trials the seat of the trouble should be located and the correct remedy found. Aside from the head-end steam line, the diagrams appear to be very good, rather above the average.

S. HURLEY.

Scranton, Penn.

In the February 22 number, C. C. Hooper asked what caused the poor diagrams. The high-pressure cylinder seems to be doing very good duty; good valve setting is shown. The receiver pressure looks to be very high. It seems to be about one-eighth of the boiler pressure, which would mean 12½ pounds for a boiler pressure of 150 pounds. This makes a very high terminal pressure in the high-pressure cylinder. I would advise him to try four pounds receiver pressure by lengthening the low-pressure cutoff. This will cause the high-pressure valve to cut off earlier and result in a lower terminal pressure and consequently a lower back pressure.

It looks as if some meddling had been done to the valve gear of the low-pressure cylinder. The admission is very late and, since it is a tandem-compound engine, this shows that the cutoff rods may have been changed so as to make the cutoff occur almost at the beginning of the stroke, thus causing the high receiver pressure. What might cause the same effect is an excessive lap of the steam valves of the low-pressure cylinder. This makes them late in opening. If the eccentric is advanced to increase the lead as on a cross-compound engine, the high-pressure diagram will be distorted.

I think the thing to do is to centralize the wristplate and see that the steam valves are given about ⅛ inch lap. Of course, the dashpot rods will have to be changed in length to suit this change, otherwise the valves would not pick up. I have tried this way of increasing the lead by decreasing the lap and have met with good results. It will be seen that the valve has to travel a shorter distance to open the port when the lap is small than it does when the lap is large, consequently the engine gets steam quicker which is what the low-pressure diagram shows that it needs. At the same time, this will not interfere with the compression.

Mr. Hooper states that the engine seemed to be working very well even when the defective diagrams were taken. This is not surprising as the diagrams show that only a normal load is being carried and the engine apparently has a large surplus of power from which to draw.

A. C. WALDRON.

Revere, Mass.

Engineers' Wages

The writer of the editorial in the issue of February 8, under the heading, "Can You Beat It?" struck the nail on the head when he said that engineers work longer hours and for a lower wage than men in other lines where they are protected by unions. I cannot help expressing my appreciation of this editorial.

We are trying, here in Des Moines, to better the engineer's condition. The wages of the operating engineer compare very unfavorably with those of skilled laborers. This does not seem equable when it is remembered that the engineer has a position of trust and great responsibility.

Engineers are frequently called upon to work overtime. They very seldom get any pay for overtime work while union workmen in many cases get double pay for any overtime work even if it is not more than an hour.

I think it is high time that our engineers' organizations paid serious attention to this matter. I think it would be an excellent thing to have the subject thoroughly discussed in the columns of POWER and hope to hear from others telling of what has been done and what in their opinion is yet to be done.

E. E. EDWARDS.

Des Moines, Ia.

An Engineer's Experience in Mexico

In Mexico City, ten months ago, I secured my present position. I was to be master mechanic, chief engineer, chief electrician and goodness knows what else. For all of this I was to be paid the magnificent salary of $375 (Mexican currency) per month. The general manager told me that they could not get the work out of the shop, the engines were shut down for half the time, running expenses were too high and the coal bill was enormous. I found things just as he represented them to be.

Well, after arriving here, I learned from the directors that they needed a man, but that I would have to show what I could do. If I showed them that I was a good man, everything which they had promised would be given to me. But while I was showing them, I was only to get $250 per month (Mexican currency).

I had gone to so much expense in getting here and thinking that here was a good place in which to crawl up the ladder at least two steps, I decided to accept their proposition.

I bunched all the operating expenses and was given a clerk to make up the figures which were turned over to the directors. Then I went to work.

First, I went at the boilers. Some of the flues or tubes had ⅝ inch of scale in them. As a result, they had been putting in two or three tubes in each boiler every week.

I had the boilers cleaned as thoroughly as possible, stopped up all air holes in the boiler settings and showed the native fireman how to fire. I then secured three old tanks and with carbonate of soda commenced to see what could be done with the boiler-feed water. I hit it right the first time, for instead of cleaning out the boilers every week we needed to clean them only once a month. Boiler cleaning had cost them $60 per week; I reduced it to $13.75. Instead of feeding cold water, I fed hot water. The average amount of coal used had been 376 tons per month; I used on an average of but 299 tons. I put the engines in good shape; the oil bill was reduced just one-half. Motors, air compressors and, in fact, everything was put in the best condition possible.

After doing all this, I put it on paper and handed it in with a request for them to keep their word. Do you know what the "Old Man" told me? He said that I was receiving the standard wages here for a machinist, that he did not see how he could possibly pay more and that he wanted a technical graduate to take charge of the shops.

The graduate is here now, and it was not more than an hour ago that he asked me if I did not think that I could cut down the oil bill. We use five gallons of cylinder oil per seven days on three 11 and 17½ by 24-inch compound condensing engines and four Worthington steam pumps. Everything runs day and night except Sundays, when only one engine and one pump are run.

This graduate asked me the other day why I blew steam off from the boilers; did I have too much?

This man is paid $400 per month and when the bonus was given out at New Year's, I got $250 (Mexican) while he got $500, who had only been on the job five weeks.

What is the use of my putting it up to the "Old Man"?

I have received orders that all reports must be turned over to Mr. M. E. I am the one who had to work night and day even to get the job. I will have to continue to do so if I want to hold it. Experience which has cost me many an hour of hard work, bother and expense must be turned over to a man who does not know how to use it or benefit by it.

I have worked for engineers who were men of long experience; it was a pleasure to work for them and I am proud to be able to say that I have references from such men.

When one commences to "put it up to the 'Old Man,'" he had better find out first what kind of a man he is dealing with.

The best schools do not turn out the morally best men.

I am looking for another job for the reason that to remain here, I will be compelled to sign up for three years at the same rate of pay.

E. MOORE.

Dinamita, D'go, Mexico.

Engine Cylinder Can Act as Pump

There has been discussion from time to time of the above subject.

The question is, will the cylinder act as a pump, or in other words, will the cylinder of a steam engine under any circumstances draw water back from the

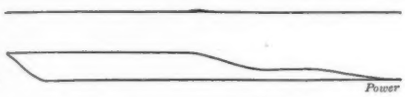


FIG. 1. LOW-PRESSURE DIAGRAM WITH NORMAL VACUUM

exhaust pipe, and thereby cause the engine to be wrecked?

Under normal conditions this could hardly be expected, for in the steam cylinder we first admit steam during a part of the stroke after which it is cut off, and during the remainder of the stroke the steam in the cylinder is expanded until it has a pressure somewhat higher than that in the exhaust pipe; then the exhaust port opens and this steam is exhausted. Under this condition, it is evident that no water will be taken back when the exhaust port opens, for water will not flow from a low to a higher pressure.

Suppose, however, that the cutoff occurs early enough in the stroke so as

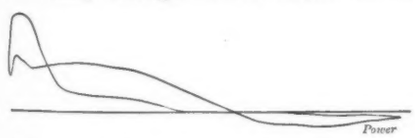


FIG. 2. LOW-PRESSURE DIAGRAM WITH VACUUM BROKEN

to expand the steam in the cylinder to a pressure lower than that in the exhaust pipe. It is certain, under these conditions, that upon opening the exhaust port there will be a "suction" from the pipe to the cylinder, and unless there is a drop in the pipe as it leaves the cylinder, any water which might be present in the exhaust pipe near the cylinder will undoubtedly be drawn back into the cylinder, and could, in the opinion of the writer, cause just such an accident as was described in a previous number.

Fig. 1 is a diagram from a low-pressure cylinder taken with a small load, while that shown in Fig. 2 was taken from the same engine under the same conditions except that the vacuum had been broken. In the latter case the expansion line drops about two pounds below the back-pressure line and when the exhaust port opens the steam and water in the exhaust pipe will be drawn back into the cylinder until these two pressures equalize themselves, as shown where the back-pressure and expansion lines cross.

In conclusion, just suppose that an engine was running under the conditions shown in Fig. 1 and the vacuum was accidently lost; if there is a rise in the exhaust pipe as it leaves the cylinder and if no provision has been made for draining, what would prevent the engine from being wrecked? Absolutely nothing.

J. M. ROW.

Fort Monroe, Va.

Responsibility of Engineers

In a recent issue of POWER, O. M. Downey told of an accident which happened in a power plant, and for which the engineer in charge was not in any way responsible. In spite of this, however, the engineer's license was revoked. This incident seems to illustrate a serious defect in the State license law, which in justice to operating engineers should be corrected.

If an operating engineer is under the orders of a master mechanic, it seems entirely just that in case of accident the master mechanic should be held equally responsible with the engineer if he is the one who is directly responsible as in the case referred to by Mr. Downing. It seems to me that this is a very good point for engineers in this State to get together upon and to agitate an amendment to the license laws to cover it.

C. C. HARRIS.

Springfield, Mass.

Cutting Oil Cup Glasses

Oil-cup and water glasses cost money and the sooner one spends seventy-five cents to buy a cutter and one or two spare wheels the more money he will save.

I have tried the string method and the sulphur match one as well as that involving the use of a file, but lots of glasses were spoiled. Perhaps I did not know how. I have always taken delight in giving a man a glass and asking him to cut it for me in his way. The result was failure in almost every case.

In cutting a cup glass using a cork washer it is necessary to cut it perfectly square or else grind it afterward. The best way in which to cut them true is to make a short scratch or cut with a cutter on the inside in one place and not over ⅛ inch from the end. Then lay the glass down carefully and in five minutes it will split off absolutely square. The reason for this is that excessive cooling strains seem to be present in all of the common kinds of glasses.

One maker of water glasses claims to avoid this defect by fusing two tubes into one. If one tries to make the mark ¼ inch from the end, the glass will not split because it will have spring enough to relieve the strain. In this case, the glass must be marked clear around and then chipped and ground smooth.

V. L. BALLOU.

Shirley, Mass.

In response to the inquiries of C. D. C. and Charles F. Chase, I will say that I have tried cutting oil-cup glasses by saturating a cord in kerosene and tying it tightly around the glass. My experience is that it will cause the glass to break under or very close to the cord, but I have always been unable to make the break even enough. In several attempts, where the glass was thick, I failed to cut it at all.

I have cut several glasses very satisfactorily by first taking a gage-glass cutter and setting the slide to extend the cutting wheel to the desired distance from the end of the glass and running the cutter around on the inside of the glass in the same manner as is employed in cutting a water glass. After having cut the outside of the glass in the same manner, I applied the cord and lighted it. The glass would instantly break at the lines traced by the cutter.

JOHN C. PITTS.

Cherokee, Okla.

In the March 1 issue, C. F. Chase wants to know what success others have had in cutting glasses by the burning of a string around them.

I have tried this method many times but never could get the glasses to break where the string had been tied. I have heard others say that it can be done, though, so there must be something in it.

The best way I know in which to do it is to wet the glass with turpentine and scratch around it with a three-cornered file. Then give the glass a few light taps around the scratch and a neat even break will result.

A rough edge on a glass can be smoothed off by wetting a file in turpentine and rubbing it over the edge.

E. GRANFIELD.

Boston, Mass.

Charles F. Chase's article on cutting oil-cup glasses recalls the method we use for cutting lubricator and water-gage glasses. It is only practicable where the glasses are only an inch or so too long.

The method is as follows: Moisten the head of a match until it becomes soft enough to adhere to the glass. Trace a line with the match around the glass both inside and out at the place to be broken. When the brimstone has dried heat the glass in the flame of an ordinary

candle or lamp. The glass will break just where the match has been applied. When heating, keep the glass turning, as it should be kept at a temperature as nearly even as possible in order to insure an even break.

ALFRED WOOLCOCK.

Gile, Wis.

With reference to recent inquiries in POWER as to how to cut oil-cup glasses, I wish to state that by observing the following rules any person of ordinary intelligence will, after a few trials on old bottles or jars, be able to cut cylindrical glass tubes without failure:

Tie a piece of woolen yarn, using only pressure enough to retain it in its position, once around the bottle at the place where the cut is to be made, and trim off the loose ends.

Select a place where there is no draft.

Have a pail full of clean cold water.

Pour kerosene over the woolen yarn around the bottle.

Ignite the kerosene-soaked yarn quickly, while holding the axis of the bottle in a horizontal position, and at the same time keep the bottle revolving, until a ring of fire is formed about the bottle.

Then steep the bottle with its axis in a vertical position into the water. That part of the bottle below the woolen yarn will then drop off.

R. E. ANDERSEN.

Brooklyn, N. Y.

Piston Ring Troubles

I read with interest an article about piston-ring troubles in the February 15 issue. Several years ago, we experienced trouble with the piston rings on the high-pressure side of a cross-compound engine. Ordinary split rings were used, but would break very soon. A two-piece sectional ring was substituted, but this did not give satisfaction. In these rings holes were drilled to admit steam behind the ring and force it into place.

The trouble was remedied in the following manner: The cylinder was rebored and a new solid piston cast, but instead of the usual snap rings, two sectional rings were used. These were in six pieces and were pressed out by flat steel springs. On each side of these rings a band of red metal was pressed on to the piston and turned to about 0.031 inch smaller than the cylinder.

I do not agree with the gentleman who makes the statement that it does not matter about the piston rings being perfectly tight. If they leak at all, it is sure to create pounding, and in large engines this is very annoying.

The rings are put there to seal the two compartments of the cylinder, and if they do not do their work something will go wrong. Not only will it affect expansion and waste the live steam, but it will also affect the compression, for the latter leaks by as quickly as the former.

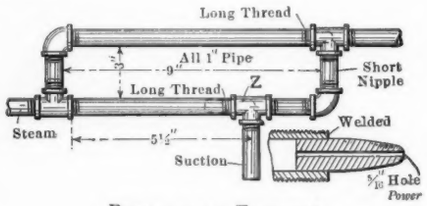
I argue that springs in piston valves wear the steam chest, and recently heard of a case where an engineer tried to run his engine without them, but his results were poor.

G. B. LONGSTREET.

Somerville, Mass.

Homemade Ejector

I notice in the March 8 number that Mr. Willard would like to have the details of construction and the dimensions of the homemade ejector which I described in the January 25 issue. Also, that Mr. Dolphin rather doubts that the ejector as illustrated would work satisfactorily.



DETAILS OF EJECTOR

The details of the ejector are given herewith. At the time when I described this device, I was under the impression that there was but one nozzle used. After reading Mr. Dolphin's criticism, however, I looked the matter up more carefully and found that two nozzles were employed, the second being located as shown at *Z*.

W. O. REGAN.

New York City.

Checking Shafting for Alinement

In the February 15 number there was an article under the above heading. The method shown in Fig. 1 for determining if a pulley is set vertical is not capable of giving satisfactory results. The surest way to check the pulley is to use a level on both the shaft and the pulley itself. The best method in which to check one shaft with another is to use a rod; a tape or a line will not be found accurate enough.

It is a fact that a receiving pulley can be made to guide a belt where desired, assuming the driver and receiver to be located exactly in line. If a belt is inclined to run on one side of a pulley it is customary to move the pulleys one way or the other, and then if this does not make the belt run right, a piece is nailed onto the pulley to make it do so. This is wrong. Certain mechanical facts should be taken into consideration. If the belt were laid out in a perfectly straight line and the pulley rolled along the belt, it would roll just as far away from the center line of the belt as a screw of the same pitch would carry it.

The remedy is to change the shaft to such a position relative to the line of the belt that the receiving pulley would run in a straight line along the whole length of the belt. On narrow fast-running pulleys we often find a belt running zigzag from one side to the other. This is caused by the two pulleys being a little out of line. By moving one a trifle the belt will run perfectly true.

J. H. HERMANCE.

Hot Springs, Ark.

The Use of Metal Cement

My attention was called to the letter on page 407 of the issue of March 1, in which the use of metal cement is recommended.

The man who called my attention to this letter claims that such a job is unsafe, and declared that he would not work near a joint where the threads were cut away as illustrated.

I fully agree with him and will add that any man who would set a trap like that, or permit others to do it ought to be forever prohibited from having charge of repairs in a steam plant.

W. H. WAKEMAN.

New Haven, Conn.

In the issue of March 1, H. Jahnke describes two methods of using metal cement in stopping leaks in piping. The first method is all right providing the holding capacity of the threads has not been impaired by rust or otherwise.

The second method can hardly be considered a safe one. The cutting away of the metal in the ell around the pipe weakens the joint, especially if the pipe is screwed into the ell for a distance of only three or four threads as is often the case. Elastic metal cement is all right for stopping leaks where it is not subjected to any very great tension.

Mr. Jahnke states that the leak was successfully cured by filling the groove with the metal cement. Admitting that the leak was successfully stopped in this way, was the job, as a whole, a successful one? Will not the cement give away under the influence of expansion and contraction which is very powerful at some ells depending on the length of the pipe, the way the line is constructed and the changes in temperature caused by shutting off the steam and turning it on? Are not the few remaining threads very liable to let go? A waterhammer would be a bad thing near or at this ell.

If the pipe had a long thread and was well screwed into the ell, the job might be safe. But long threads well screwed in do not usually leak.

A better way to repair such a leak would be to use a pipe clamp, of which there are several designs on the market. When one is properly applied it is very effective and does not weaken the joint.

JAMES W. BLAKE.

New York City.

English Practice in Condensing Equipment

The Latest Type of Jet Condensing Plant. Some Notes on Air and Circulating Pumps, Giving Recent Types, and Systems of Electric Drive

BY JAMES A. SEAGER

The latest type of jet-condensing plant which is being introduced into English power houses, in order to meet the demand for very high vacua due to the increasing use of the turbo-generator, is the Leblanc patent system. In Great Britain this is controlled by P. J. Mitchell, of Westminster, London, S. W., and is being received with considerable favor. For example, at the power house of the Clyde Valley Electrical Power Company, one set dealing with 50,000 pounds of steam per hour, and other examples could be adduced. These are constructed either with a simple jet for capacities up to 500 kilowatts or multiple jet for the larger sizes. Fig. 1 shows a sectional arrangement of the simple-jet form, an integral part of which is the Leblanc rotary pump to be described later. The steam enters at the top of the opening and meets the cooling water ejected at a very high velocity from the pelton wheel shown. This, together with rapid condensation, sweeps the condensed steam out against atmospheric pressure, at the same time creating and maintaining the desired vacuum.

This type of plant is always arranged in such a manner that it has to lift its injection water. It is therefore necessary to provide some means of priming the pelton wheel. A steam-starting ejector is arranged for this purpose, though, if a supply of water under pressure is available, this steam ejector is not necessary. With the usual type of ejector condenser there is always a risk of water going back into the engine cylinder, but with the Leblanc simple-jet condenser, this is impossible, because, immediately the vacuum fails, the water supply is cut off, whereas, with the old type of plant, the water pours into the condenser from the pump, whether there is vacuum or not so long as the pump is running. The efficiency of any type of condenser depends upon the removal of the air entrained with the steam and injection water. In the injector type of condenser the jets of the water are continuous, so that the removal of the air depends upon the friction of the jets on the air.

With reasonable velocities of jets the maximum amount of air that can be removed is one volume of air to one volume of water. On the other hand the discharge from the pelton wheel in the Leblanc simple-jet condenser is intermittent, a sheet of water being discharged from each blade in the wheel, so that the air is entrapped and removed in a positive manner. By this means from six to ten volumes of air are removed to one volume of water. The efficiency of the Leblanc simple-jet condenser is, therefore, very high and its capacity for dealing with air leaks (which have a very deleterious effect on the vacuum in ejector condensers) is also very great.

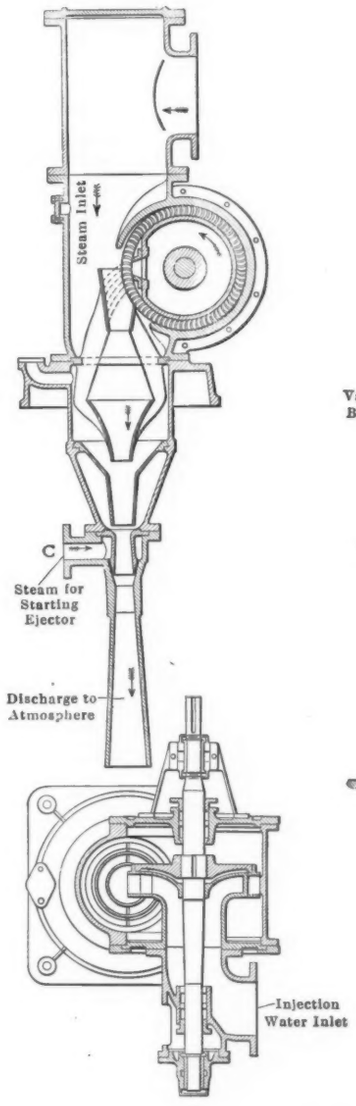


FIG. 1. LEBLANC SIMPLE-JET CONDENSER

It occupies very little floor space, requires practically no foundations, has low power and water consumption, requires practically no attention when running, and can be driven by belt from the engine flywheel or direct coupled to an electric motor. The maintenance is also extremely low, and there are no reciprocating parts or valves to be renewed.

The multiple-jet condenser, shown in Fig. 2, differs only in the method of mixing the injection water and steam, and extracting the condensate. The steam to be condensed is led into the top of the condenser at *A*. The injection water enters at *B* and passes into the distributing chamber *C*, from which it is discharged into the condenser through the nozzle *D*. These nozzles are of large area and are fitted with vanes which split up the jets of water and impart to them a swirling motion. By this means the injection water presents a very large surface for contact with the steam. Lower down in the condenser a cone *E* is fitted, in passing through which the steam and

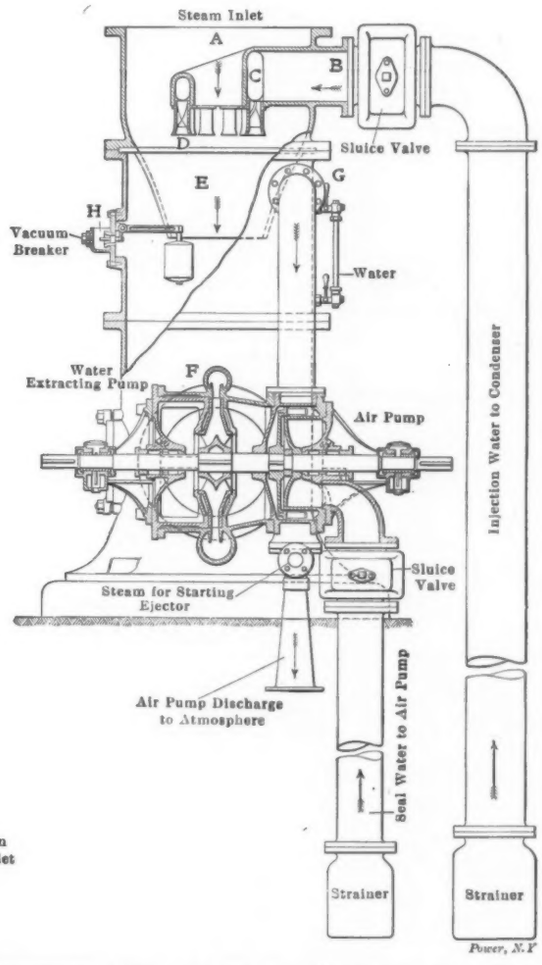


FIG. 2. LEBLANC MULTIJET CONDENSER

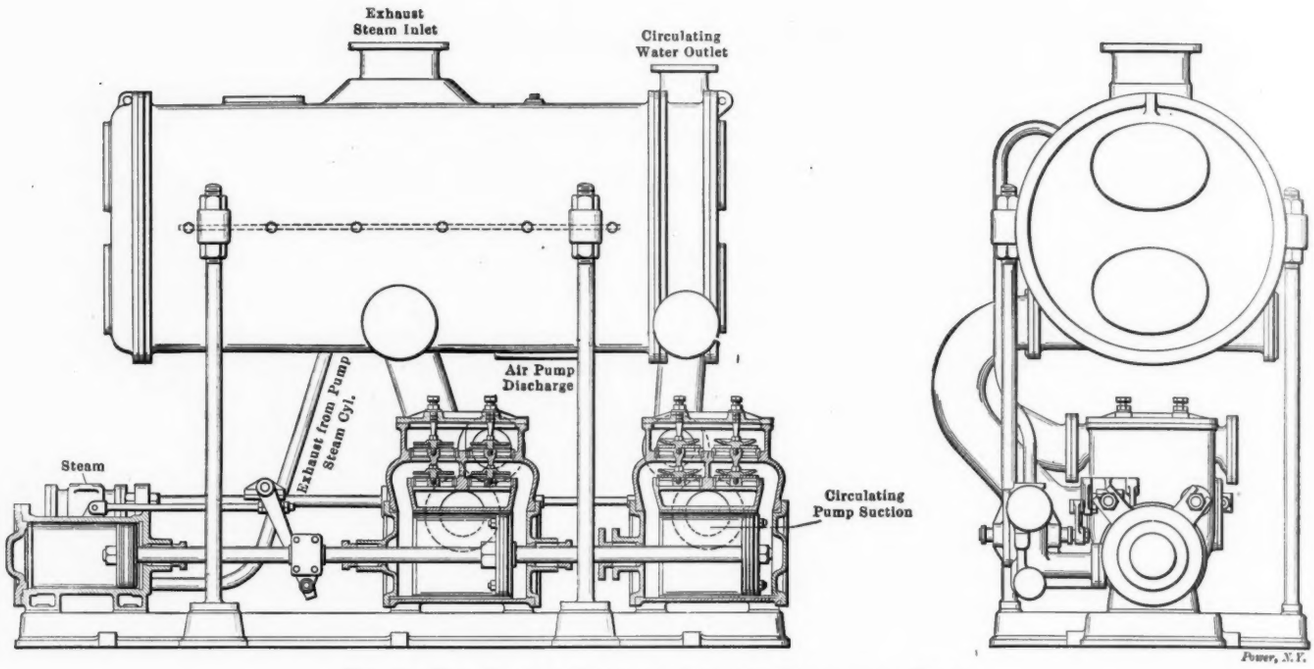


FIG. 3. THE CLARKE-CHAPMAN SURFACE CONDENSING PLANT

injection water are thoroughly mixed, so that condensation takes place very rapidly and completely. When the injection water enters the condenser it has a considerable velocity due to the difference between the atmospheric pressure and the pressure inside the condenser, and in passing through the cone *E* this velocity is utilized to give the air and incondensable gases a first compression. The condensed steam and injection water fall to the bottom of the condenser after passing through the cone *E* and are removed by the water-extracting pump *F*. The air and incondensable gases separate from the water and are drawn up to the air-pump suction pipe at *G*.

The water-extracting pump is of the centrifugal type, specially designed to draw the condensed steam and ejection water against a high vacuum. It can also be arranged, if necessary, to discharge the water straight to a cooling tower. The air pump is of the Leblanc rotary valveless dry-air type. The impellers of the water-extracting and air pumps are mounted on a common shaft and placed in the same casing. This arrangement makes the plant very compact, and requires the use of only one motor to drive both pumps. There is a clear and uninterrupted way through the pumps, so that, should the pump stop for any reason, air rushes into the condenser through the air pump and immediately breaks the vacuum. In addition there is an air valve on the condenser controlled by a ball float, which rises with the water when it gets above a certain level. This opens the air valve and admits air to the condenser, and so breaks the vacuum.

This type of plant is specially designed for giving the high vacuum necessary for turbine work. The head room required is small (being less than 9 feet in the case of 1500-kilowatt sets), so that the condenser can generally be placed immediately below the turbine exhaust branch and the full vacuum obtained on the turbine blades. This is of great importance, as with many types of plant—the barometric jet for instance—the loss of vacuum between the condenser and the turbine may be as much as 0.5 inch. The power absorbed by the pumps under normal circumstances is a very small percentage of the output of the engine or turbine exhausting into the condenser. The pipe connections required are exceedingly short and simple, and the construction of this type of plant is such that very dirty water may be used for injection, as the apertures through which the water has to pass are all amply large, nor are there any valves to choke up nor tight-fitting pistons to score or seize.

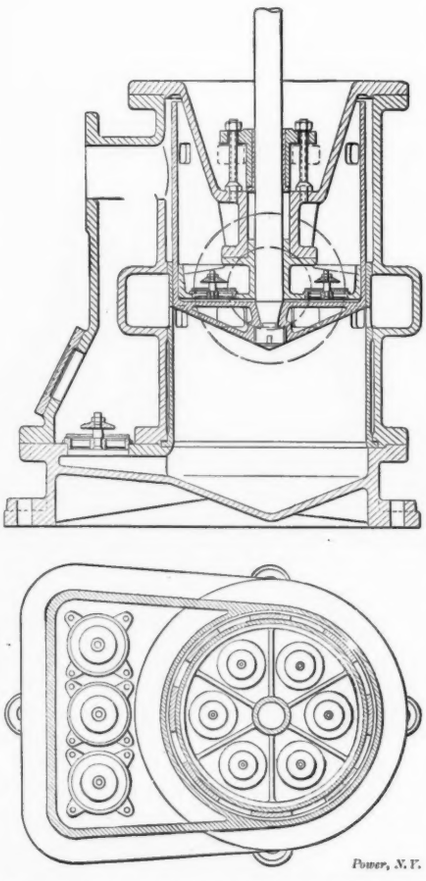


FIG. 4. ZYLBA DOUBLE-ACTING AIR PUMP

AIR AND CIRCULATING PUMPS

The subject of condensing is so intimately connected with questions of the removal of air and condensed water that this survey of modern practice in power-house condensing plants would not be complete without some reference to air and circulating pumps as at present used. Fig. 3 shows a type which may be called the standard practice in many plants, especially where marine-type reciprocating engines are employed. This particular plant was made by Clarke, Chapman & Co., Ltd., of Gateshead-on-Tyne, and embodies Woodeson's patent pumps. Both the air pump and circulating pump are of the reciprocating type, steam driven, and for the condenser illustrated, having 1350 square feet of cooling surface, the steam cylinder is 12x15 inches, the diameter of the air pump and circulating pump being both 14 inches. That such pumps are efficient in performance may be judged by the examples in the accompanying table of Clarke-Chapman plant installed in British power houses.

The usual limit for this type of plant appears to be sizes dealing with 25,000

to 30,000 pounds of steam per hour. For large sizes and high vacua now obtaining in power houses of magnitude it has been found advisable, in the case of air pumps, to discriminate as far as possible between the portion of the plant dealing with the condensed steam and that which abstracts the included air. For this reason, as will have been noted, provisions are made to draw the air in many cases from the condensers from a different point (preferably the coolest) of the condensing chamber to that forming the outlet of the condensate, and separate pumps are installed, the dry-air pump dealing with the former and the wet pump with the latter.

Recent Pumps

Quite recently the Midland Engineering Company, Ltd., has introduced into power-house practice a new pump, known as the "Zylba," which is of the suction valveless type similar to the Edwards, but made double-acting by the arrangement shown in Fig. 4. The position of the bottom delivery valves makes them all accessible by a single inspection door, while the top delivery valves are got at through a door on the top. These pumps have been supplied to several corporations, to the British war office, Italian government, etc., and have proved exceedingly satisfactory. There is practically no friction between the sliding bucket and the liners, owing to a suitable clearance being allowed for expansion and contraction due to temperature variations. The effective water seal prevents any air leakages back into the pumps, and

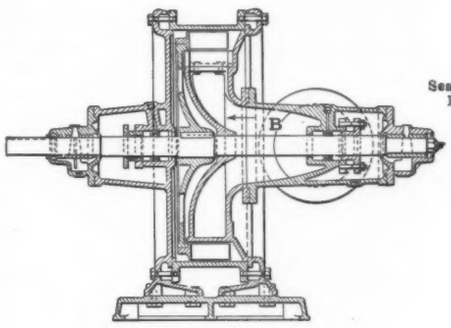
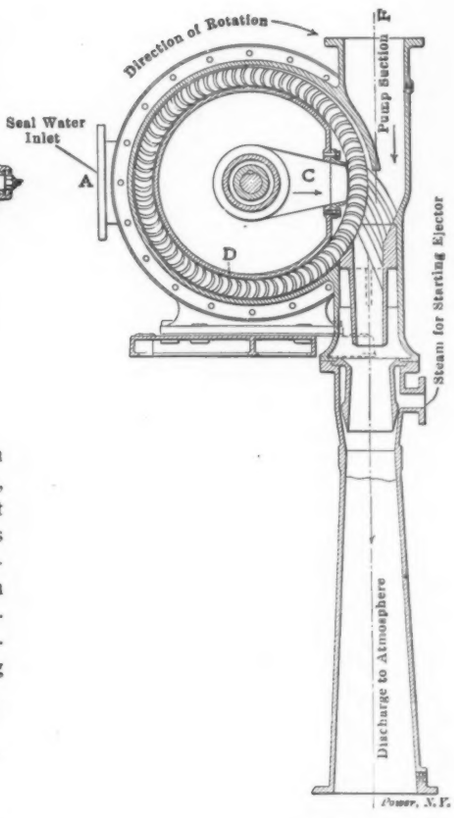


Fig. 5. Leblanc Air Pump



PERFORMANCE OF CLARKE-CHAPMAN CONDENSING PLANTS.

Lb. of Steam per Hour.	Condensing Surface Sq.Ft.	Vacuum, In. of Hg.	Pumps.
6,000	700	27 to 28	Air and circulating, Chapman 10″x12″; 8″ steam cylinder.
12,000	1350	27 to 28	Chapman, 13½″x15″; 12½″ steam cylinder.
10,000	1000	27 to 28	Chapman, 18″x15″; 8″ steam cylinder.

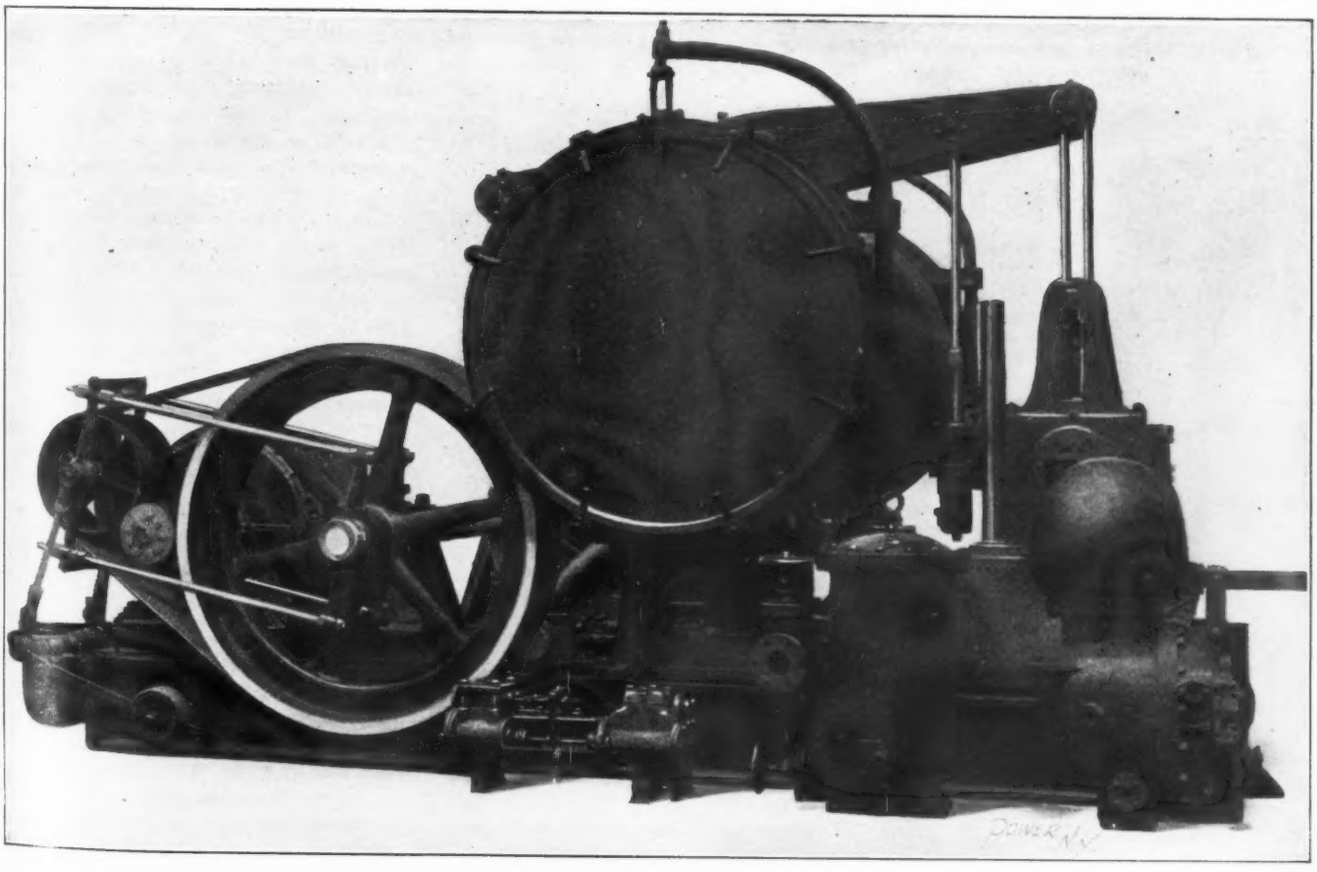


Fig. 6. Electric Drive for Medium-size Condensing Plant

the vacuum obtainable in these pumps is 29.9 inches with the barometer at 30 inches measured by a mercurial gage. The pumps, being double-acting, have the danger of water shock and consequent breakages of the pump is entirely eliminated. The pumps can run at high speeds if necessary, although for usual can be reduced by one-half for a given duty, which fact naturally reduces the cost of upkeep and amount of attention required.

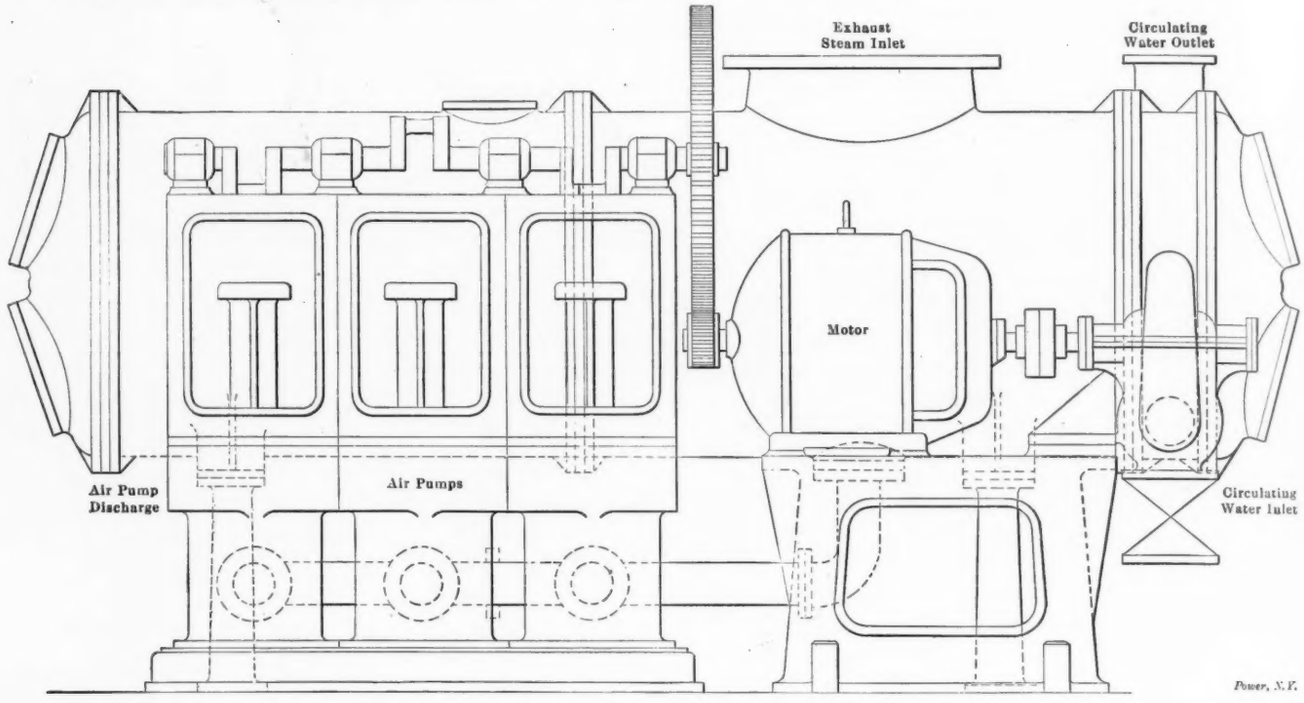


FIG. 7. COMPACT DRIVE ADOPTED BY ISAAC STOREY & SONS

a perfectly balanced up-and-down stroke, and can therefore be used with success as single-crank pumps. When the sizes are large, the pumps are built in two- or three-crank units and by setting the cranks at 90 to 120 degrees respectively, the torque obtained is exceptionally even. power-house purposes the speed is kept within 150 revolutions per minute. Where the driving motor runs at high speeds, the power is transmitted to the pump by means of machine-cut double helical gearing. It is obvious that the pump, being double-acting, possesses twice the displacement of a single-acting pump of the same size and speed; the number of pumps, and consequently working parts,

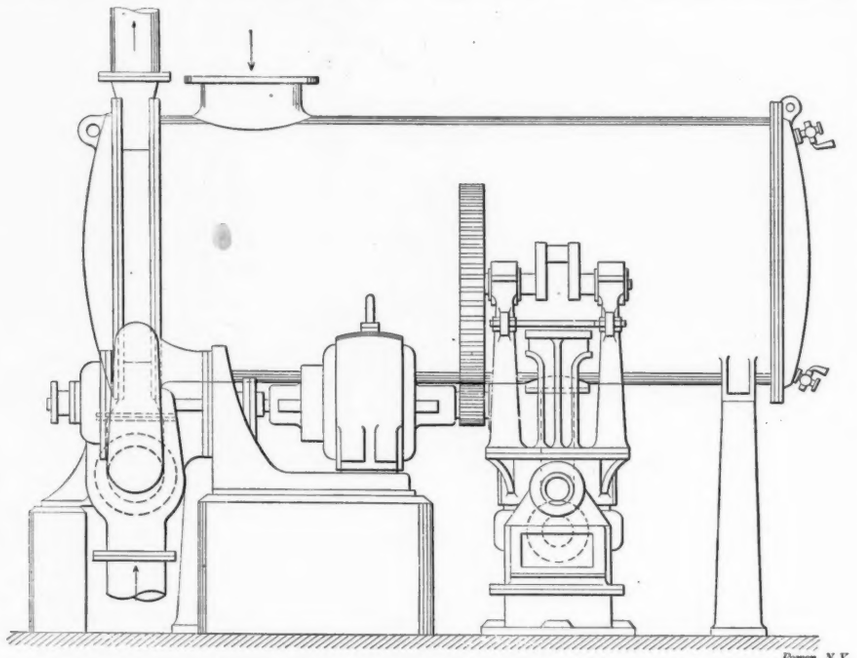


FIG. 8. ELECTRIC DRIVE USED BY MIDLAND ENGINEERING COMPANY

The bottom delivery valves in the pump are ready to receive any water that may collect in the pump and pipes so that

Reference was made above to the Leblanc air pump. This is shown in Fig. 5, and is a valveless dry-air pump, consisting primarily of a reversed pelton turbine wheel in conjunction with an ejector. Sealing water is introduced through the branch *A* into the central chamber *B*, from which it passes through the port *C*. It is then caught up by the blades *D* of the pelton wheel, which is rotated at a suitable speed, and ejected into the discharge cone in the form of thin sheets having a high velocity. These sheets of water meet the sides of the discharge cone and thus form absolutely tight water pistons which entrap the air and incondensable gases coming from the condenser through the suction branch *F*, and carry them out against the atmospheric pressure. In passing through the pump, the sealing water receives practically no increase in temperature. The same water may therefore be used over and over again, or it may be passed through the condenser as cooling water after it has passed through the air pump, without lowering the efficiency of the plant.

As this pump is of the rotary type, it is especially suitable for direct coupling to a high-speed engine or electric motor and for being run at a very high speed. The space and foundations required are therefore reduced to a minimum. There are no reciprocating parts, valves, waste, clearance spaces, or pistons to wear and cause leakage; consequently, its efficiency is high, the attention required when run-

ning and its maintenance being negligible, and there is no loss of efficiency even after long service. This type of pump has been applied with satisfactory results to low-level and barometric-jet, surface and evaporative condensers, and on account of its high efficiency, the quantity of cooling water or the cooling surface may be reduced to produce the required results.

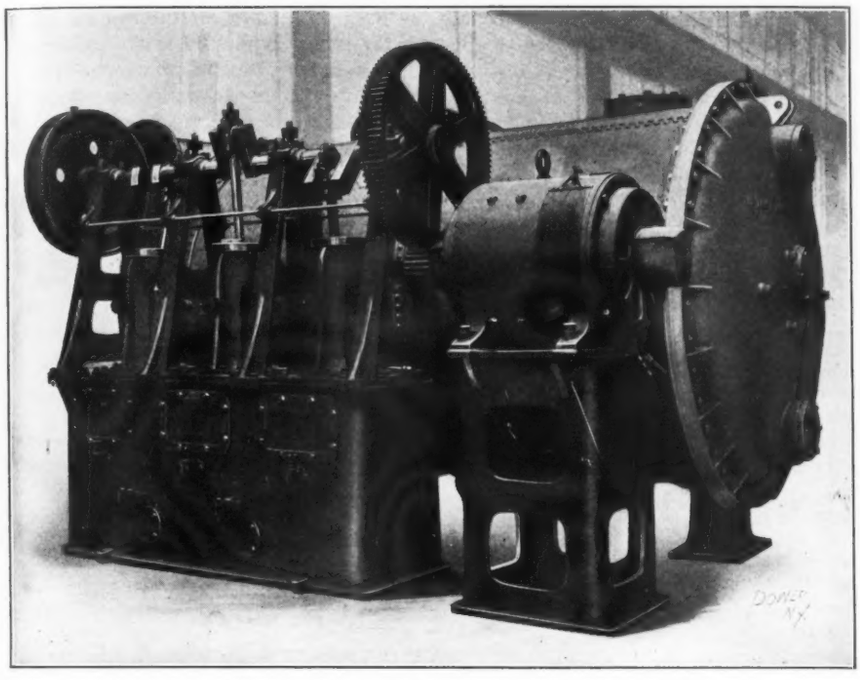


FIG. 9. SEPARATE MOTOR RESERVED FOR COOLING PUMP

ELECTRIC DRIVE FOR PUMPS

It is only to be expected that in electric power houses considerable use is now being made of electric motive power to drive the auxiliary machinery of the generating station, although for some time considerable hesitancy on this point was experienced by British engineers on account of the disastrous effect on the system of a total shutdown of the main station, should there be no alternative means of setting the station to work again. Nowadays, however, reliability of electric supply has reached such a high pitch that motors are commonly used to drive both the air pumps and circulating pumps for condensing plants attached to the main sets, and Fig. 6 shows one method of arranging such a drive which has been adopted for a medium-sized plant in the London area. This is a condenser of 1150 square feet cooling surface made by Easten, Anderson & Goolden, Ltd., which firm is associated with The Pulsometer Engineering Company, Ltd., of Reading. The pumps are driven through a Wimshurst & Hollick reduction gear; from the first motion of which is driven an Edwards air pump, and from the second motion a deep-well three-throw circulating pump. The circulating water consists of dirty Thames water, which at times contains a mass of floating chips so that special arrangements are made for the easy removal of the covers for cleaning out the tubes. Fig. 7 shows a compact drive adopted by Isaac Storey & Sons, Ltd., by means of which one motor operates both the centrifugal circulating pump and the three-throw air pump, the difference in speed required by the two sets being obtained by means of a spur on to the air pumps, while Fig. 8 shows the same type of drive used by the Midland Engineering Company, Ltd., of Birmingham, for operating its Zylba double-acting air pump and a circulating pump. As in most power houses the amount of space available for condensing plant is small, extreme compactness of arrangement is required, which is evidenced in this plant.

Fig. 9 shows a larger condensing plant, having a cooling surface of 6000 square feet, and triple air pumps of 12-inch diameter by 14½-inch stroke. This was supplied by the Brush Electrical Engineering Company, Ltd., of Loughborough, for a 1000-kilowatt turbine set under very stringent specifications as to cooling water temperature, and it will be seen that a separate motor is reserved for the circulating pump. Fig. 10 shows the type of direct-driven pump supplied by this firm for circulating water through a condenser of 2260 square feet cooling surface; the pump is 9 inches in diameter at the orifice, and is capable of delivering 1200 gallons of water per minute.

It will be seen from the foregoing survey that English condenser practice in electricity-generating stations is at present particularly interesting as it appears to be passing through a stage of transition. The advent of the turbo-generator has brought in its train a demand for a degree of vacuum which was not before necessary, and the various devices mentioned to increase this vacuum are worthy of very careful attention. The returning favor of the jet condenser in its improved forms is a somewhat remarkable feature, and the growing confidence on the part of English engineers in placing their auxiliary plants under the control of electric power generated by the main sets is a pleasing indication of the reliability of electric power supply. It is therefore hoped that the above notes will be of value to engineers as a résumé of what has been done up to the present time in the design and construction of the condensing plant.

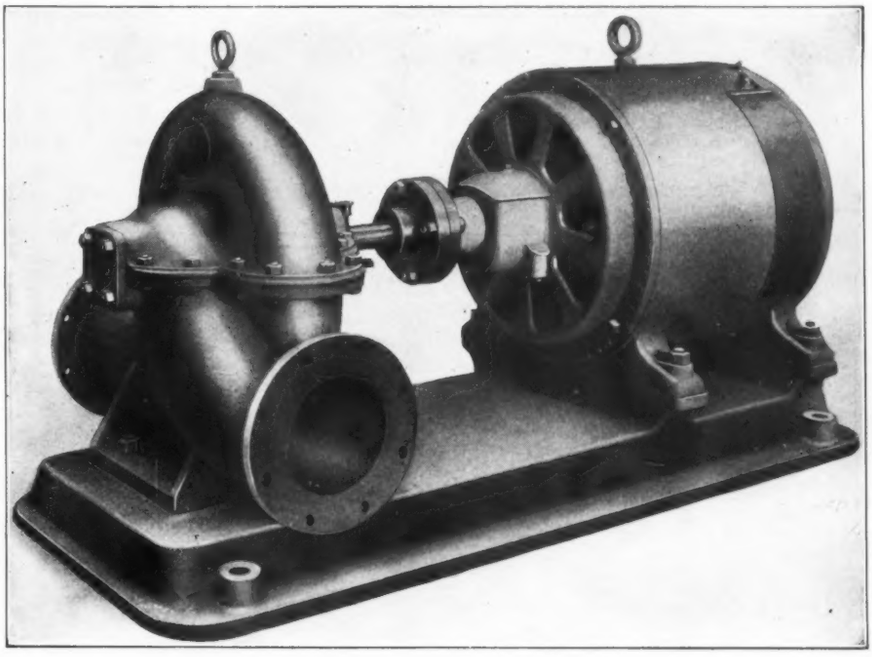


FIG. 10. BRUSH DIRECT-DRIVEN PUMP

An 8000 Horsepower Superheater

Duty Is to Superheat Steam from 8000 Horsepower of Boilers by Means of Waste Heat from Four Billet Heating Furnaces

A large superheater was installed and put into operation during November, 1909, in one of the plants of the United States Steel Corporation, by the Power Specialty Company, of New York. The duty of the heater is to superheat the steam from boilers having a capacity of 8000 horsepower, by means of the waste heat from four billet-heating furnaces. A short description of the installation will be of interest, because of the large size of the unit and the unusual conditions of service.

The superheater is of Foster construction and is made up of 306 elements or superheater tubes 12 feet long and arranged vertically in seven banks, one behind the other, giving an arrangement of 18 elements deep by an average of 17 elements wide. At the top the elements are expanded into steel return headers and at the bottom into wrought-steel connecting headers. There are 14 of the latter, seven connected to the inlet manifold and seven to the outlet manifold, so that the steam passes from the inlet manifold to the inlet connecting headers, through the elements and return headers and to the outlet manifold. The arrangement of the elements is such that the temperature of the steam leaving the first bank, although coming in contact with the hottest or entering gases at

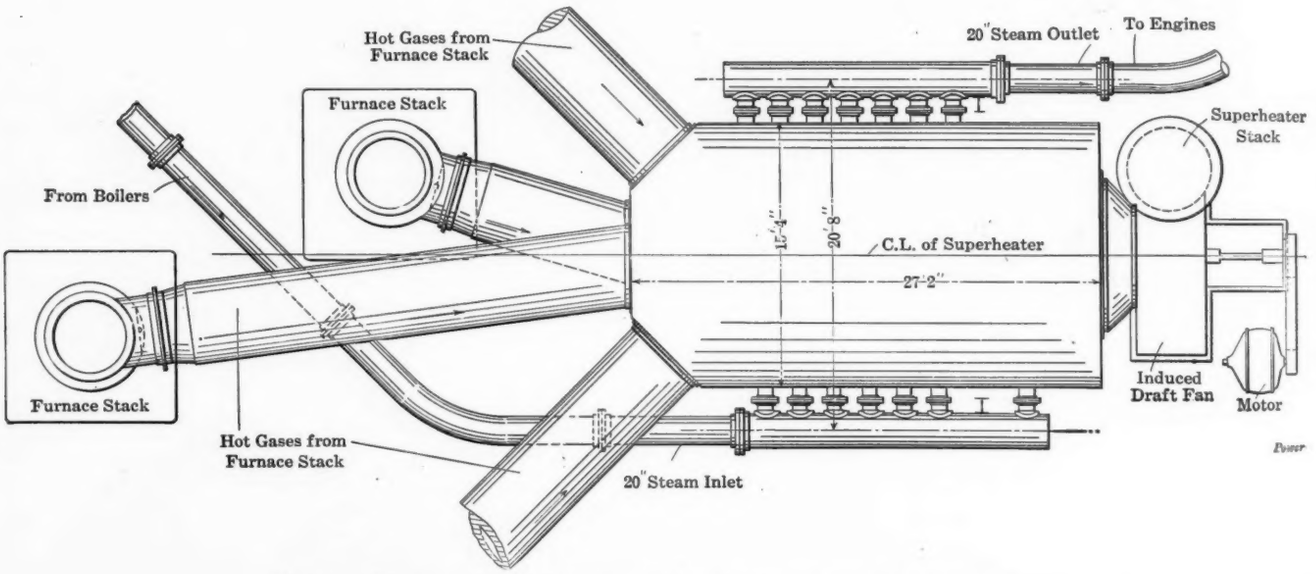


FIG. 1. GENERAL ARRANGEMENT OF SUPERHEATER AND CONNECTIONS

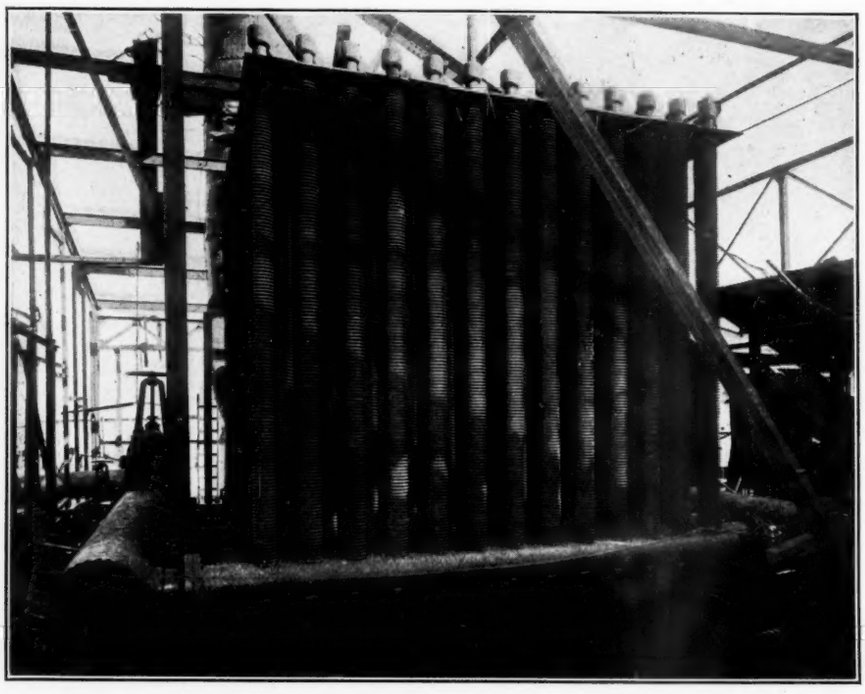


FIG. 2. SUPERHEATER DURING ERECTION BEFORE SETTING WAS IN PLACE

about 1250 degrees Fahrenheit, is closely the same as the temperature of the steam leaving the last or seventh bank, where the temperature of the gases has been reduced to 450 degrees as they leave the superheater. The last bank is arranged on the countercurrent principle, thereby insuring a low temperature in the gases leaving the superheater. This arrangement of surface insures the thorough distribution of the work over the entire surface and avoids concentrating the work on the first few rows of elements.

Fig. 2, from a photograph taken during construction, shows the heating surface before the setting was in place. The setting is made of paneled steel plates, lined with asbestos blocks and 4½ inches of firebrick, with an air space between the asbestos and firebrick. The panels are removable, giving easy access to all parts of the interior. Soot-blowing and cleaning doors are provided. The superheater is erected on a platform supported on columns, over the billet-heating fur-

naces from which the waste gases are supplied.

Fig. 1 shows the general arrangement of the superheater and connections. The hot gases are received through four brick-lined flues from the four stacks of the billet-heating furnaces and pass through the superheater, which is arranged with the inlet and outlet manifolds at the bottom, one on each side. The gases are drawn through the heater by means of an electrically driven induced-draft fan, providing easy control of the temperature of the steam leaving the superheater. The induced-draft fan is regulated to pull the gas necessary to heat the steam passing through the superheater the desired amount, the balance of the furnace gases going up the furnace stacks.

Steam is supplied to the heater from three different boiler houses containing Stirling boilers and having an aggregate capacity of 9520 horsepower. The steam from about 8000 horsepower of these boilers goes through the superheater, where it is superheated 100 degrees to a final temperature of 470 degrees Fahrenheit. The contract required that 240,000 pounds of steam per hour at 145 pounds pressure, containing 3 per cent. of moisture, should be superheated 100 degrees corresponding to a final temperature of 463 degrees Fahrenheit.

The superheated steam is used in the following equipment, consisting of the main engines together with the usual number of smaller engines, pumps, etc., to be found in a steel mill:

One 42¾ and 75 by 66-inch Porter-Allen compound engine, driving a rod mill, one 34 and 54 by 66-inch engine which has a Corliss high-pressure cylinder and a Porter-Allen low-pressure cylinder, driving a continuous rod mill; and two Nordberg engines, one 40 and 68 by 60 inches and one 30 and 52 by 48 inches, driving rod mills.

The big Porter-Allen engine is about 250 feet away from the superheater, and with a temperature of the steam at the superheater of 470 degrees, the temperature at the engine throttle was 425 degrees. The receiver pressure on this engine was 30 pounds, and the temperature of the steam in the low-pressure steam chest was 274 degrees, showing that saturated steam was supplied to the low-pressure cylinder with about 75 degrees of superheat at the throttle. At some of the engines nearer to the superheater, where the temperature is higher and where the cutoff is later, resulting in a higher receiving pressure, the steam to the low-pressure cylinder contains from 20 to 30 degrees of superheat.

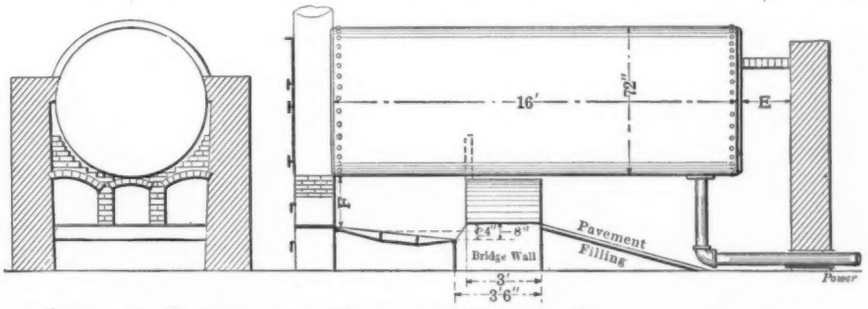
A Smokeless Soft Coal Furnace

BY T. T. PARKER

In a recent number of POWER, J. O. Bender seems to infer that it is impossible to abolish smoke when using soft coal in standard, horizontal, tubular boilers with common settings. If Mr. Bender had spent any time as an operating engineer in Chicago he would have run into more schemes for doing away with smoke than one would find elsewhere. The problem has to be solved there to suit the smoke inspector armed with a camera. During the several years which I passed in Chicago employed as a boiler inspector, I had an opportunity to observe a number of appliances, special furnaces, etc., designed to fit existing conditions of settings and do away with smoke. The more successful ones consisted of brick arches of various shapes placed in different parts of the furnace. Some would be located directly over the fire at mid length of the grates, while others would extend from the front of the bridgewall to the rear head.

To spring an arch from side wall to side wall is easy enough, but its durability is quite short. As a result of tests and trials, the three-arch plan shown in the figure was evolved and with careful firing gave excellent results. The figure shows a standard 72-inch boiler, 16 feet long. There are two unknown dimensions, *E* and *F*. The most common value for each of these is 24 inches. Of course, where *F* is 36 or 42 inches, the problem of smokeless combustion would be easier to solve, as we would have a larger combustion chamber over the grates wherein the volatile matter has a chance to become ignited prior to the gases passing over the bridgewall.

Confining ourselves to the common-sized settings and referring to the figure, it will be noted that the grates are dropped 4 inches at the rear end. The bridgewall should be at least 36 inches thick on top although 42 inches is better. On the top of the bridgewall two piers are erected as shown. They should be built of the best firebrick. These piers should be at least 36 inches long. From the piers the arches are sprung as shown. The center arch should just clear the shell of the boiler. If key-shaped arch brick are obtainable, they should be used. The space between



STANDARD SETTING WITH THREE-ARCH TYPE OF COMBUSTION FURNACE

arches and shell should be filled in with bricks laid without mortar joints to a distance from the shell of about an inch.

Behind the bridge a good combustion chamber should be built of the shape shown. It is a matter of fact that the cleaning out of a combustion chamber is often neglected and this impairs the efficiency of any furnace. To get results, this chamber should be cleaned at least every two weeks.

In starting up a cold boiler having this kind of a furnace, let the night gang get up steam, for the furnace will certainly smoke until the arches get hot. After this takes place, a fairly good fireman will eliminate 85 to 90 per cent. of the smoke, providing the boiler has a fairly heavy load. Of course, it is understood that a hot fire is desirable and if a boiler is not worked hard, the grate surface should be reduced proportionally by bricking off the bars at the bridgewall end. If the fuel used is mine-run, the large lumps should be broken up so as to maintain an even rate of combustion on each square foot of grates. In regard to the admission of air above the grates, I believe that better results are obtained by admitting it at the lower edge of the fire doors rather than elsewhere. Some people perforate the deadplate for this purpose. I favor an aperture of from 3 to 4 square inches, divided between the two doors, for a 72-inch boiler with a draft of ½-inch, water gage, when using this furnace. A ratio between grate and heating surface of 1 to 45 or 50 gives good results, but 1 to 55 will give better.

I desire to call attention also to a very valuable feature of the furnace. The cold air is prevented from striking 95 per cent. of the heating surface when the fire doors are open for firing and

cleaning fires. If the arches had no other value, if smoke was not objectionable or if smokeless fuels were used, the arches would prove of value in giving up heat when cold air is rushed through the furnace. To maintain an even, high temperature is important from every viewpoint. Contractions are reduced and tubes hold tight and wear much longer.

With regard to the distance from the rear head to the back wall, I will say that this is but too often left to the judgment of a mason. Many boiler shops advise 16 to 18 inches as the proper distance. This is, of course, all wrong. If this space is restricted, the velocity is increased and the upper tubes do most of the work, thus causing the lower ones to leak. I have seen a great deal of trouble with leaking tubes cured by setting back the rear wall to 30 inches and better evaporation was secured too, as the gases were more evenly distributed to the tubes.

In some rare cases conditions are such that steam jets are necessary with this furnace, but in most cases excellent results are obtained without them. In every case which came under my observation, the evaporation per pound of coal was increased. The soot deposited at the rear end was found to be of a grayish color instead of the usual black. This indicated that combustion had been completed by the time the gases had passed through the arches. No evil effects were found on the shell over the arch. Indeed, with a ½-inch shell and a girth seam over the bridgewall, this furnace affords excellent protection to the seam, minimizing the danger of trouble from fire cracks.

There will be found a slag on the floor of the bridgewall which should be chipped off from time to time. Of course, the baffles on top of the arch are only one brick thick.

This furnace, practically handled, delivers the goods with ordinary settings and I heartily recommend it to Mr. Bender and others. The cost of installation is very low, about $25 per boiler. What may be done to abolish smoke is another story.

The Smoke Abatement Act for Boston

There has been submitted by the Fuel-supply Committee of the Boston Chamber of Commerce a "Proposed Act for the Abatement of the Smoke Nuisance in the City of Boston and Vicinity."

The territory over which the proposed act shall be effective consists of Boston harbor, the cities of Boston, Cambridge, Somerville, Everett and Chelsea, and the town of Brookline.

The act prodives for the creation of "The Metropolitan Board for the Abatement of Smoke;" classifies chimneys; outlines the manner in which the density of smoke is to be established and prescribes the degrees of smoke which shall be prohibited from issuing from chimneys of the different classes.

The Smoke Board is to consist of the chairman of the Massachusetts State Board of Health, *ex officio*, and four other members to be appointed by the governor, with the advice and consent of the Council, to serve a term of four years each. The first appointments, however, are to be for one, two, three and four years respectively, so that there will be retired, each year, only one member of the Board. One member of the Board is to be a merchant or manufacturer, one a lawyer, one an engineer, and one a representative of the Public Service Commission.

The Board is to appoint a secretary and a chief smoke inspector and as many deputies and assistants as may appear necessary. The salary of the secretary is to be $2500 and that of chief smoke inspector $3600 per year. Other salaries are to be fixed by the Board. The members of the board are to receive no salary, but any personal expenses which they may incur while engaged in official duty are to be repaid by the State.

The duties of the Board are to enforce the provisions of the smoke-abatement act; to investigate complaints of violations of the act; to order the observance on the part of persons or corporations of the provisions of the act and to enforce such orders by legal proceedings.

Chimneys are classified by the act as follows:

Class 1 includes all fixed or stationary stacks having an inside area at the top not exceeding the area of a circle 5 feet in diameter.

Class 2 includes all fixed or stationary stacks having an inside area at the top greater than the area of a circle 5 feet in diameter but not exceeding the area of a circle 10 feet in diameter.

Class 3 includes all fixed or stationary stacks having an inside area at the top greater than the area of a circle 10 feet in diameter.

Class 4 includes all stacks of vessels having an inside area at the top not exceeding the area of a circle 4 feet in diameter.

Class 5 includes all stacks of vessels having an inside area at the top greater than the area of a circle 4 feet in diameter.

Class 6 includes all stacks on steam locomotives.

The density of smoke issuing from a chimney is to be estimated by means of the Ringelmann charts which are used by the United States Geological Survey. In the table is given the number of the smoke chart which corresponds in appearance to the most dense smoke which, under the act, may be emitted from the chimneys of the various classes together with the lengths of time for which such emissions are permitted to continue.

It is proposed to have the act take effect on June 1, 1910.

MAXIMUM DENSITY OF SMOKE AND LENGTH OF TIME FOR WHICH IT MAY BE EMITTED.

	Class of Chimney.												Locomotives Moving Trains of Six Cars or More.	
	I		II		III		IV		V		VI			
During the Year.	Chart No.	No. Minutes during One-Hour Period.	Chart No.	No. Minutes during One-Hour Period.	Chart No.	No. Minutes during One-Hour Period.	Chart No.	No. Minutes during One-Hour Period.	Chart No.	No. Minutes during One-Hour Period.	Chart No.	No. Seconds in 5-Minute Period.	Chart No.	No. Seconds in 5-Minute Period.
1910	3	6	4	5	4	10	4	9	4	12	3	40	3	50
1911	3	4	3	10	3 including 4	20 including 5	3	12	3	15	3	30	3	40
1912	2	8	3	6	2 including 3	30 including 10	3	7	3	9	3	20	3	30
1913 and subsequently	2	6	3	3	2 including 3	25 including 5	3	3	3	5	3	20	3	30

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
355 Dearborn Street, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for March, 1910, *was* 178,000.

April 5	36,000
April 12	37,000
April 19	37,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

	PAGE
Norfolk's Traction Power Plant	702
Feed-water Heaters of the Closed Type	706
Four 9000-horsepower Water Turbines	708
Engine, Turbine or Combination	711
Common Diseases of Boilers	712
Power Loss of a Slipping Belt	713
Primer of Electricity	714
Inexpensive Alternating-current Station Recording Watt-hour Meters	716
Repairing Holes in Commutators	717
Gas Engine Ignition	718
Three years' Record from a Diesel Engine Plant	718
Tests of an Exhaust-gas Water Heater	720
Gas Power Investigation for the Navy	720
Practical Letters: Water in the Cylinder....Packing Water Piston of Small Pumps....Testing Boiler with Hot Water Meter....The Gyroscope....A Method of Cooling Brake Pulleys....Why Do Not Engineers Use the Slide Rule?....Automatic Compensating Bearing....A Novel Feed-water Heater....Fittings for Superheated Steam....The Calorimeter....Trouble with Water Supply System....A Choked-up Drip....Engine Troubles....Homemade Torch....Removing Piston Rod from Crosshead....Engine Troubles....An Automatic Lubricator	721-726
Discussion Letters: "Freak" or "Fake" Diagrams?....Engineers' Wages....An Engineer's Experience in Mexico....Engine Cylinder Can Act as Pump....Responsibility of Engineers....Cutting Oil Cup Glasses....Piston Ring Troubles....Homemade Ejector....Checking Shafting for Alinement....The Use of Metal Cement	727-730
English Practice in Condensing Equipment	731
An 8000-horsepower Superheater	736
A Smokeless Soft Coal Furnace	737
The Smoke Abatement Act for Boston	738
Editorials	739-740

Power and The Engineer

Editorial

State Boiler Inspection Department for Ohio

A bill to provide for the better protection of life and property against injury or damage resulting from the operation of steam boilers is under consideration in Ohio. This bill is backed by the representative operating engineers of the State, a copy of it having been sent to each of the State N. A. S. E. associations; all but two it is said, indorse it unqualifiedly.

The bill provides that the department be placed in charge of the chief examiner of steam engineers, which latter official is to receive $1800 a year in addition to the salary now allowed him by law for acting in this dual capacity. The bill provides, furthermore, that he shall have two assistant chief inspectors at a salary of $2400 per annum each, a chief clerk at $1800, and such other inspectors of boilers at $1500 each and assistant clerks at $1200 per annum each, as may be necessary—provided the total expenses of the department do not exceed a variable maximum provided in the bill. Necessary office and traveling expenses incurred in the discharge of their duties are allowed to each in addition to his salary.

The bill provides that all steam boilers and their appurtenances in the State of Ohio, "except boilers of railroad locomotives, boilers under the jurisdiction of the United States, and boilers in private residences, shall be thoroughly inspected internally and externally" at least once a year. Furthermore, the hydrostatic test may be applied if deemed necessary. For this inspection service, a fee of $5 is charged for each internal and $2 for each external inspection. All owners or users of such boilers must report to the chief inspector annually the location of their boilers.

It is claimed by the advocates of the bill that Ohio has become the dumping ground for condemned boilers from the State of Massachusetts and its purpose is to stop this traffic and also the indiscriminate selling of second-hand boilers by junk dealers, who, with a fresh coat of black paint may doctor up an old condemned boiler and dispose of it to innocent purchasers. Instances of this practice are cited and it is pointed out that Ohio, which is now generally looked up to and respected for its State license law, is too modern and progressive a community to allow such practices to continue.

Compulsory boiler inspection is the only method by which all the unsound boilers in a community can be reached and the public safeguarded, as it is unfortunately true that the very people who will buy without question a second-hand boiler will, from motives of economy, refrain from having said boiler examined by a competent boiler-insurance company.

The plan of exempting boilers that are insured from State inspections is pursued in Great Britain, in Massachussets, and in Connecticut, and in the cities of Omaha, St. Louis and Philadelphia. It would undoubtedly be of advantage to Ohio to follow the same plan, but legal advice on this question has brought out the fact that in Ohio this arrangement would be unconstitutional and that every steam user would of necessity be compelled to undergo State inspection. This will increase the steam user's boiler expense per year and in just so much will it arouse his opposition.

We have not seen the decision upon which the allegation of unconstitutionality is based. The point involved is probably the inability of the State to delegate its powers to insurance companies. This is effected in Massachusetts by making the inspectors of the insurance companies virtually a part of the inspection system by requiring them to be examined by the inspection bureau and certified before their inspections will be accepted by the State. If the Ohio bill can be amended in some way so that double inspection and two fees will not be necessary, a serious objection to its passage will be obviated.

It is refreshing to learn that one of the large insurance companies of the country is working hard for the bill, and is lending its support to legislation which might readily turn to its disadvantage. Such action is to be highly commended, for only too often the reverse course is pursued.

Ohio has been a leader in the West in the matter of the license question and the effort to establish State boiler inspection is another step in the right direction, which will be watched with interest by all engineering bodies. Other States should get in line.

Hydroelectric Power for Central Stations

Such exaggerated ideas have been spread through the country broadcast in recent years concerning the value of hydroelectric power, that it is timely to consider the simple facts which should determine the purchase or rejection of such facilities by lighting and power stations which happen to lie within the distribution or transmission radii of water-power installations. Boiled down to its final pith, the problem is simply one of reliability of the offered supply and the total yearly cost of it as compared with the cost of local generation. This total yearly cost must in each case include fixed charges as well as operating expenses, of course. Each case which the engineer or station manager is called upon to consider will contain different numerical factors, but in all instances the controlling factors are the two just mentioned.

Reliability, in central-station work, is of an importance not easily exaggerated, and as the size of the plant grows, with increasing demands upon it by its community, continuity of good service transcends all other considerations. We are not intimating that lack of dependability is common with hydroelectric companies; we merely wish to emphasize the economic fact that service reliability simply cannot be overlooked by a light and power company serving a city of even the tenth order of importance. Nothing in the way of low price can offset the embarrassment and losses which interrupted or poor service entails. The transmission line is a very vulnerable link in the chain of hydroelectric service and, hence, until more accurate data are made public concerning the interruptions of such systems, it is supremely important for the central station taking power from a hydroelectric system to provide for some sort of emergency service, either in the form of reserve capacity in local generating machinery or in auxiliary capacity in the transmission lines.

The cost of furnishing service compared with the purchase price necessarily involves a careful estimate of the interest, depreciation, taxes, insurance and sinking-fund requirements upon the investment necessary to handle the business, added to the complete cost of operation, including fuel, labor, repairs, supplies, legal expenses and any running costs which may apply in the local situation. In some cases these figures can be obtained without expert assistance outside the company; in others, not. The most difficult question is that of depreciation, but if the company has kept careful records of the life of its equipment and the frequency with which different classes of apparatus become obsolete, certain limits may be fixed upon which experienced judgment may act. It is important to realize that although energy can often be generated about as cheaply in a well organized steam- or gas-power plant as it can be purchased from a water-power company, taking merely the operating cost into account, the addition of the fixed charges to the operating cost in many cases will turn the scales in favor of the hydroelectric service contract.

Where reliable service is obtainable, and a certain amount of reserve capacity can be counted upon, it is a great advantage for a central station to be relieved from the necessity of planning for additions to its generating plant for a long term of years. On the other hand, if the service offered is likely to be somewhat uncertain, due to the unreliability of the transmission organization's system or to wide variations in available power which may occur at the hydraulic generating plant, it may be a better plan to install generating equipment for a relatively large proportion of the load, making use of the water-power service as a convenience and an auxiliary supply instead of as a mainstay.

A Commendable Practice

It has long been observed that the majority of boiler explosions happen in the morning just after the load has come on, frequently on Monday morning, after the boilers have passed through a period of idleness. During this period they have been under the care of watchmen and helpers who, of course, are not experienced engineers. The night watchman, having orders to have steam raised at a certain time in the morning, brings the pressure up regardless of any engineering precautions which, with the regular men in charge, are observed as a matter of course; and owing to the lack of exhaust steam during the early morning hours the feed water is frequently pumped into the boilers cold while the fires are being pushed to get steam up to the prescribed point before the day watch comes on.

There is no doubt that this firing-up time is the crucial point in the daily experience of a steam generator, which repeated day after day must surely have an influence on its structure, and if the truth were known it would probably be surprising to see how much abuse the average boiler stands in its lifetime.

It is a pleasure, therefore, to cite the practice of one engineer who appreciates the extent to which unequal expansion and contraction in firing up can cause trouble in boilers. This man is of the opinion that engineers do not always realize the long time it takes to get the water in a boiler heated to a uniform temperature, even in those boilers which are generally supposed to have good circulation. He is in charge of a battery of boilers in which he had noticed that as long as one hour after raising steam to one hundred pounds, the working pressure, cold water could be drawn from any one of the bottom drums, which would be cool enough to drink. To avoid this condition, before firing up he would raise the water level one gage higher than required when running, and when five or ten pounds of steam showed on the gage, he would blow down to the regular level. This emptied the lower drum of all cold water and immediately brought the boiler and contents to a uniform temperature, after which steam could be raised as rapidly as desired. This is a little part of routine management that has fully repaid the trouble taken to keep it in force, as not one cent has been expended on these boilers since they were put in.

This engineer previously had charge of a battery of return-tubular boilers fitted with mud drums. The practice above outlined was used here also and in thirteen years not a tube had to be expanded nor a cent expended for repairs on these boilers.

The most careful and intelligent handling is necessary to get the best results with any boiler and it is difficult to overdo this feature; in fact, it is better to spend a couple of minutes a day in being too careful than to spend a lifetime of regret over not being careful enough.

"In every great engine and under every big boiler you'll find a square fire box. Well—we have adapted that principle to our furnaces," advertises the maker of a house heater.

We did not know that there was any especial merit in a square furnace, and had rather assumed that the locomotive and ordinary stationary furnace took the rectangular shape because it would be more trouble to build them round. For boilers with a circular plan like the Corliss, Manning, etc., round furnaces are used with entire satisfaction.

In making engineering calculations be careful to keep the units consistent; for instance, never multiply feet by inches. One of the most frequent mistakes of this nature occurs in computing the indicated horsepower of an engine by the formula,

$$PLAN \div 33{,}000.$$

In this case the mean effective pressure P is always given in pounds per square inch; consequently, the area A, upon which it acts, should be given in square inches in order to obtain the total pressure upon the piston. The 33,000 is in foot-pounds per minute, hence the length of stroke L should be in feet and N in strokes per minute. In using this formula how many of us have paused to wonder whether L should be in feet or inches and N in strokes or revolutions?

INQUIRIES of General Interest

QUESTIONS are NOT ANSWERED UNLESS they are ACCOMPANIED by the NAME and ADDRESS of the INQUIRER.

Size of Pipe to Supply Radiators

What should be the diameter of the main steam pipe to supply 1000 square feet of radiators with steam at 212 degrees and the temperature of the surrounding air at 70?

A. D. B.

With a temperature difference of 142 degrees each foot of radiating surface will give off 255 heat units per hour and as each pound of steam in condensing to water at 212 degrees will give up 966 heat units, 264 pounds of steam per hour will be condensed. As steam at atmospheric pressure weighs 0.038 pound per cubic foot, 6970 cubic feet of steam per hour or 1.94 per second will be required. At a rate of flow in the steam pipe of 25 feet per second the area of the pipe would be 0.077 square foot or 11.1 square inches, which corresponds very closely with the area of a circle 3¼ inches in diameter; 4 inches is the diameter of the nearest commercial size of pipe.

Effect of Advancing the Eccentric

What effect on the port opening in a plain slide-valve engine will an advance of the eccentric have?

F. A.

None whatever. Advancing the eccentric will bring the different events earlier in the stroke, but will not affect the length of valve travel.

Hard and Soft Patches

What is the difference between a hard and a soft patch on a boiler?

P. J. F.

A hard patch is one which is riveted to the sheet and made tight by calking while a soft patch is bolted on and is made tight by packing with red lead, salammoniac and iron borings or something that will answer the purpose.

Rule for Valve Setting

Please give a short rule for setting engine valves.

A. B. R.

See that valve and eccentric rods are of such a length that there will be equal port opening at both extremes of valve travel. Then with the crank pin on the center turn the eccentric in the direction it is to revolve until the valve shows lead at the end of the cylinder corresponding with the crank pin.

Size of Steam and Exhaust Pipes

What proportion to the cylinder should steam and exhaust pipes have?

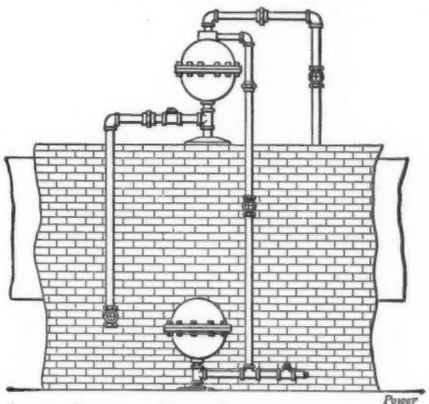
Steam-supply pipes should be of such a size that the velocity of flow will not exceed 100 feet per second. The velocity of flow in the exhaust pipe should not exceed 60 feet per second.

Duplex Trap System

What is a duplex trap system and how does it operate?

T. W. J.

A duplex trap system is one in which two traps are placed in series, one of which collects returns from heating coils



DUPLEX TRAP ARRANGEMENT

or other low-pressure sources, and discharges into another placed above the water level of the boiler. The latter discharges into the boiler by gravity when the pressure in the trap is equalized by the admission of steam from the boiler to the trap above the water in it.

Meaning of the Term "Horsepower"

State the meaning of the term horsepower.

B. C. D.

The term horsepower is used to designate the rate at which work is being done. It is the equivalent of 550 pounds raised 1 foot per second.

Saving Returns

If you took charge of a plant where they were heating a building with five pounds pressure and the returns were wasted, what would you recommend doing to save the returns if the boiler was set too high to return by gravity?

W. M.

A return trap should be installed. It might be necessary to put in two traps in series.

Safety Valve "Blow Back"

If you had a boiler on which the safety valve was set to blow off at 200 pounds and the inspector came around and ordered you to set the safety valve to blow at 160 pounds and after setting it you get 10 pounds "blow back," how would you remedy it?

B. C. D.

Pop safety valves are provided with an adjustable seat ring which by its position determines the amount of "blow back." Raising the ring increases and lowering it decreases the amount of "blow back."

Rust Joints

What materials are used in making rust joints and how are they made?

C. W. D.

Rust joints are usually made with sifted cast-iron turnings or borings, sal ammoniac and sometimes a little sulphur mixed to a stiff paste with water and packed into the joint with a calking iron. One part sal ammoniac, 2 of sulphur and 80 of iron, by weight, are given for quick setting, and 1 sal ammoniac, ½ sulphur and 100 of iron for slow setting. It is better to omit the sulphur if the job is not a "hurry" one.

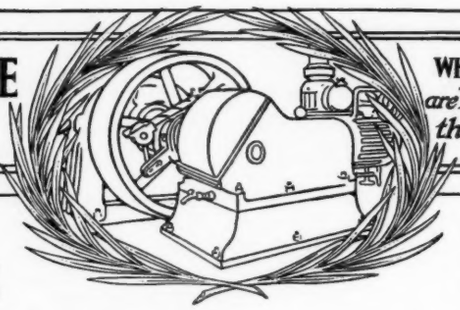
Brazing and Soldering

How are thin pieces of steel or iron, such as bandsaws and the like, brazed?

J. H.

Sheet steel or thin pieces, such as bandsaws, are generally brazed with silver solder, which is sold in strips. For brazing a bandsaw, the ends are tapered so as to form a scarfed joint. The ends are securely held in position, leaving a space all about the joint. A small piece of solder, wet with soldering fluid, is placed in the joint; then with a pair of blacksmith's tongs, heated to nearly a white heat, grasp the joint. Sufficient heat will be transmitted to the steel to melt the solder and unite the ends. The tongs should be held in contact with the saw until the solder has set, otherwise the parts will be imperfectly united.

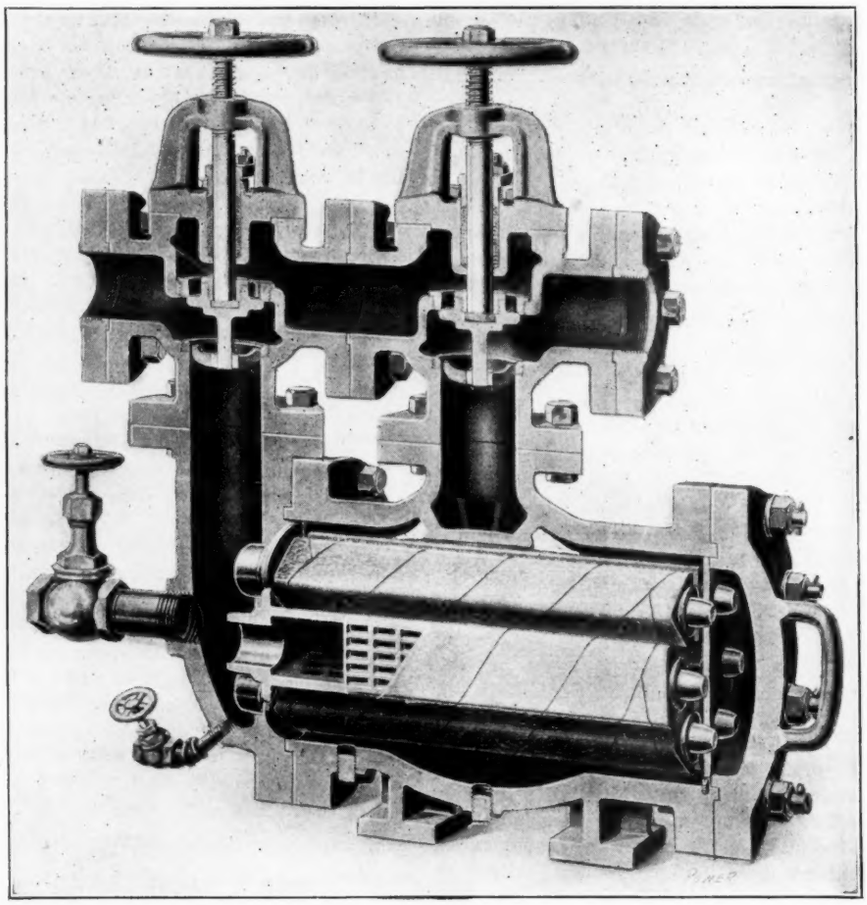
NEW POWER HOUSE EQUIPMENT



WHAT *the* INVENTOR *and the* MANUFACTURER *are* DOING *to* SAVE TIME *and* MONEY *in the* ENGINE ROOM *and* POWER HOUSE

American H_2O Filter

The accompanying illustration shows a sectional view of this filter, and the arrangement of the two controlling valves. The filter consists of a cast-iron body. The heads are bolted in place, the bolts securing the outside head being of the swing type, which facilitates the removal of the filtering cages, of which there are four. These cages or cartridges are made with three perforated sides, as shown. They are covered with Turkish toweling commonly known as linen terry. These cartridges are held in place by end projections which pass through a pressure plate and are further secured by spring washers, placed between the outside end of the cartridge and the pressure plate. The inner end of each cartridge is open for the escape of the filtered water.



AMERICAN H_2O FILTER

Bolted to the top of the filter body are two valves which control the inlet and outlet water. These valves have a double seat, so that when the filter is cut out of commission and the valve disk seated on the lower valve seat, the water is bypassed by the filter through the inlet and outlet valves. Both valves are flanged and bolted together as shown.

When the filter is in operation, the disk of each valve is opened against the upper seat. The water to be filtered then enters through the inlet valve and passes into the main filter chamber containing the filter-cartridges and after passing through the filtering material enters the outlet chamber and passes out through the outlet valve into a heater or direct to the boiler.

When it becomes necessary to remove the filtering material, the main valve disks are closed against the lower valve seat. This removes all pressure from the main filtering body. After opening the drains in the bottom of the filter body, the covers can be removed. This releases the pressure plate and the cartridge may be taken out and the filtering material renewed.

The filter cartridges can be temporarily cleaned by means of a reverse current of steam. To do so, close the main valve disks on the lower seat, open the bottom drain and the steam valve shown at the outlet end of the filter. This will deliver a reverse steam pressure to the inside of the cartridges, which will force the clogging element to the outside of the filtering cloth and out through the drain.

A pressure gage is connected to the main and outlet chambers of the filter, not shown. A water relief valve is connected to the main chamber to relieve it of excessive pressure. It is usually set at 25 pounds above the boiler pressure. The small valve at the bottom of the outlet chamber is for the purpose of flushing that section of the filter.

The filter is made by the American Steam Gauge and Valve Manufacturing Company, 208-220 Camden street, Boston, Mass.

Manzel "Class H" Oil Pump

This oil pump is so designed that all of the working parts are inside of the reservoir, and, as they work in oil, dust and dirt cannot get to them. The entire mechanism is attached to the cover, which makes it an easy matter to remove for repairs when necessary.

Two plungers are used. One plunger draws the oil from the reservoir and forces it through the sight glass; the other plunger forces the oil to the steam pipe connected to the steam cylinder. Fig. 1 shows a sectional view of this pump; Fig. 2 shows an outside view of the single-feed pump.

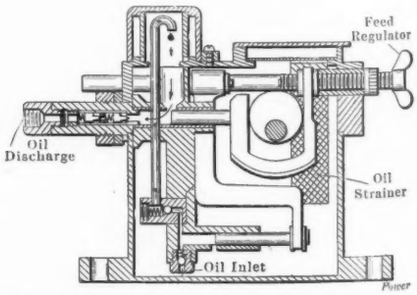


FIG. 1. SECTIONAL VIEW OF CLASS H OIL PUMP

Referring to Fig. 1 the sight-feed glass is seen conveniently placed on top of

the pump casing, where the feeding of oil can be observed from various points in the engine room. The sight glass is not under pressure, neither is a liquid used in it. Further, it can be removed while the pump is running.

The feeding of oil is regulated by simply turning the regulating screw to the right or left until the pump is adjusted

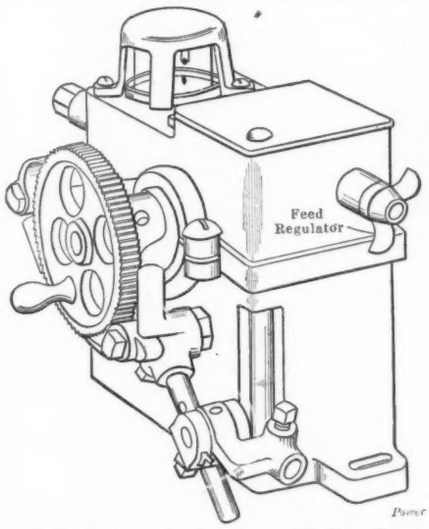


FIG. 2. SINGLE-FEED, CLASS H OIL PUMP

to feed the proper amount of oil. When once adjusted a lock nut prevents the adjusting screw from working out of place.

This type of pump is made with from one to six feeds, each feed independent of the other in the regulation of the oil supplied to the cylinder.

A hand attachment for use before starting the engine, for momentarily supplying an extra amount of oil, or for use while the engine is running, is supplied. A gage glass is also placed on the pump to indicate the amount of oil in the reservoir.

The pump is finished plain and smooth on the outside, which makes it an easy device to keep clean. It is nickel plated, and is made by the Manzel Brothers Company, 315-319 Babcock street, Buffalo, N. Y.

Electrically Operated Valve

The accompanying illustrations show two views of one design of electrically operated valve, manufactured by the Chapman Valve Manufacturing Company, Indian Orchard, Mass.

The first installation of this type of valve made by this company was in 1901 in compliance with a request made by the Boston Edison Company to equip some of their valves with electric motors. Since that time, the design has been changed to meet different conditions, so that today any type of large valve can be operated by a motor.

This motor-operated valve is easily opened and closed, regardless of the position it may occupy, and in case of an accident which makes it impossible to get to the valve and shut it, is a safeguard against loss due to escaping steam or water. As a time saver on a valve requiring opening and shutting frequently, the advantage of the motor is self-evident.

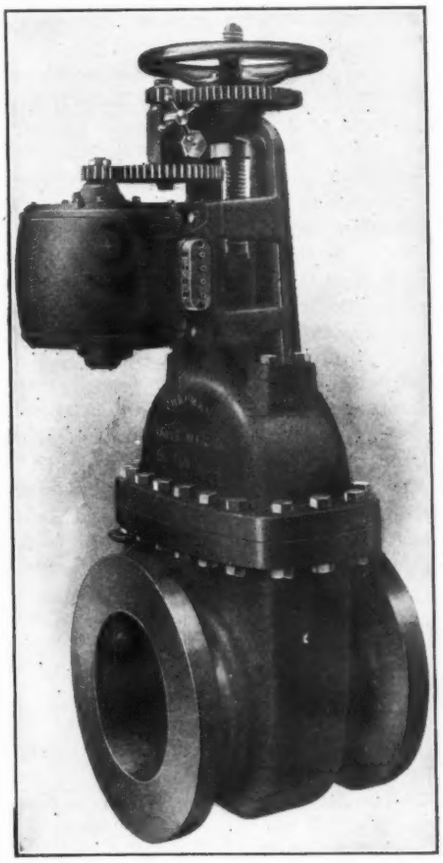


FIG. 1. GEARS IN MESH

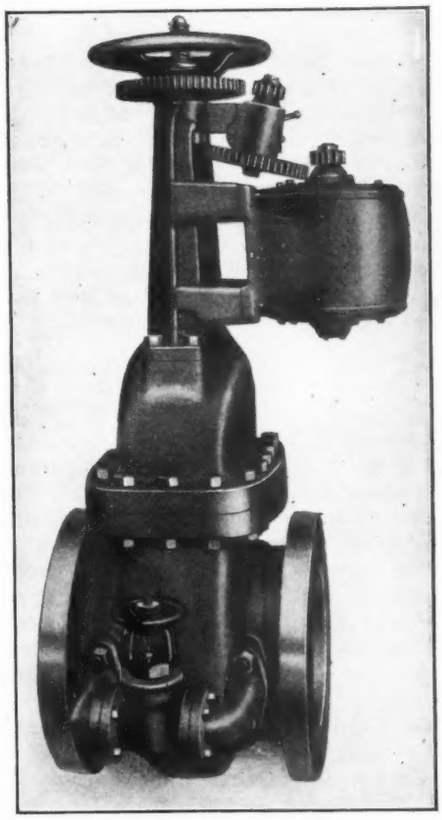


FIG. 2. GEARS OUT OF MESH

An electrically operated valve can be placed between the boiler and steam header and between the sections of header in many steam plants. This allows of any faulty section of pipe or header being isolated by merely closing the switches necessary to start the motors driving the valve stems of the valves controlling that section of the pipe or header.

A valve operated by a motor is also suitable for hydraulic systems, water works and manufacturing plants.

No contact device is used to cut out the motor at the end of the travel of the valve. The gate is forced to its seat and the motor stops. This causes the overload device on the controller to release and the handles to fly back to the out position. Both direct- and alternating-current motors are employed.

The illustration, Fig. 1, shows the intermediate gearing between the motor shaft and the valve-stem gear in mesh. Fig. 2 shows the gears out of mesh. These gears are locked in either in or out position by a locking bolt. This feature permits operating the valve by hand, should it be desired.

Museum of Mechanic Arts

A museum of mechanic arts has been recently proposed by John Riddell, mechanical superintendent, General Electric Company, Schenectady works. Mr. Riddell recently addressed the following letter to the American Society of Mechanical Engineers:

The rapid advancement in mechanic arts in this country causes machinery and mechanical devices to be quickly superseded by newer developments. Such superseded devices are often most interesting, in studying the development of engineering, but unfortunately such devices disappear, leaving no record of what has been done. Think for a moment how interesting it would be to have the original and a few intervening forms of such American devices as the cotton gin, and a few of the numerous machines which we have made for harvesting cotton, the sewing machine, telephone, phonograph, linotype, telegraph, aëroplane, and other machines, while not of American origin, but to which we have greatly contributed in their development: as the steam engine, plow, machine tools, general automatic machinery, bicycle, automobile, steam turbine, electric dynamo, and steel-mill machinery.

In the South Kensington museum the epoch-making inventions of Watt and Stephenson are carefully preserved, and visited by thousands of people from all civilized countries.

American engineers have contributed

so much to this work that it is appropriate that a record of the most important developments should be kept. I have thought for some time that a movement should be put on foot to establish a museum of mechanic arts, and that it is most appropriate that this should be done by the American Society of Mechanical Engineers.

It is important if anything is to be done in this line that action should not be delayed, since it will be readily understood that much material is available for such a museum now that could not be had ten years hence.

NEW PUBLICATIONS

TABLES OF EFFICIENCIES OF BUTT AND DOUBLE-STRAP JOINTS, ETC. Computed by W. W. Ramsay, State Inspector of Boilers, and issued by the Board of Boiler Rules of Massachusetts. 48 pages; 6x9, paper, illustrated.

This collection of tables is published and distributed by the Commonwealth of Massachusetts as an aid to inspectors authorized to pass upon the safety of boilers in use in that State, and as a help toward uniformity and efficiency. Its gratuitous distribution is therefore limited to boiler manufacturers authorized to manufacture "Massachusetts Standard" boilers, insurance companies authorized to inspect and insure boilers in Massachusetts, consulting engineers identified with the design of Massachusetts Standard boilers, members of the Boiler Inspection Department and inspectors who hold the certificate of the department. There are 84 tables which give the efficiencies of different types of butt-strap joints with tensile strength of 55,000, 57,000 and 58,000 pounds and indicate the manner of failure as well as the percentage which the joint should stand of the full plate value. Four additional tables give the Crushing Strength of Steel Plate in Front of Rivets; Tensile Strength of Steel Plate One Inch Wide, and of Different Thicknesses; Areas of Circular Grates, and Decimal Equivalents of Common Fractions.

A new bulletin has just come to hand, containing desirable information dealing with heating coils. It is gotten out by the American Blower Company, Detroit, Mich. In this catalog the maker announces that a *bona fide* actual rating as to the capacity rating of pipe coils has been given, which can be absolutely relied upon and holds good in any case where heating coils are to be installed.

A table, which appears for the first time outside of the company's own private data books, and gives the capacities of sections in terms of square feet of actual heating surface, is included in the bulletin.

SOCIETY NOTES

The National Association of Cotton Manufacturers will hold its eighty-eighth meeting on April 27 and 28, 1910, in the Mechanics Fair building, Huntington avenue, Boston, Mass., beginning on Wednesday at 11 a.m., and holding later sessions at 2 p.m., on the same day, and at 10 a.m. and 2 p.m., on Thursday. The following papers of interest to POWER readers will be presented: Buying and Handling of Steam Coals, Choice of Power for Textile Mills, Electric Drive as a Manufacturing Proposition, Progress of the Diesel Engine, Superheated Steam and Superheaters.

The Massachusetts State branch of the International Union of Steam Engineers held an open meeting in Carpenters hall, Lynn, Mass., on the evening of April 4. The hall was crowded with engineers from Lynn, Boston, Salem and other nearby cities. The discussion of the evening was on the "Value of Organization." Among the speakers invited to address the meeting were Fred L. Johnson, associate editor of POWER; William M. Beck, member of the Board of Boiler Rules, State of Massachusetts; Arthur M. Huddell, first vice-president of the International Union of Steam Engineers; Fred W. Detheridge, State president, and William R. E. Whalen, president of local No. 16.

On Monday, April 4, at a meeting, attended by nearly all of the steam engineers of the city, Birmingham Lodge No. 3, National Association of Stationary Engineers, was organized, and officers elected. The officers were installed by G. J. Gibney, of Mobile, one of the State officers of the association. G. R. Simcox was elected president and John Anderson corresponding secretary.

During the winter Mr. Simcox has devoted considerable time to calling on the engineers of the district endeavoring to get them interested in license laws and an organization to secure them.

PERSONAL

On April 1, Henry D. Shute was appointed to the position of acting vice-president of the Westinghouse Electric and Manufacturing Company. For seventeen years Mr. Shute has been associated with this company, and his promotions from time to time have been of a character to give him a broad experience in shop, sales and executive work. He studied electrical engineering at the Massachusetts Institute of Technology, from which institute he was graduated in 1892. Following his graduation, he spent a year's study in Germany at the School of Mines, Clausthal, and also in Dresden. Mr. Shute is a member of the American Institute of Electrical Engineers; a member of the Engineers' Club of New York, and most of Pittsburg's business and social clubs.

Marine Boiler Explosions

It is reported that on April 2 while the cruiser "Maryland" was on her way to Monterey bay from Santa Barbara, the tubes in one of the forward boilers blew out, terribly scalding at least three firemen and water tenders, one of whom has died. No details of the accident are available.

On April 7 one of the boilers of the "Cairnrona" exploded. The vessel, carrying 900 passengers, was on its way from London to Portland, Me., and was off Beachy Head at the time of the explosion. The accident resulted in the death of a baby and the serious injury of nine of the crew and several passengers.

Fuel Economy and Smoke Comsumption

An interesting lecture on the above subject, illustrated by lantern slides, was delivered at the rooms of the Blue Room Engineering Society at the Engineering building, Boston, on Thursday evening, April 7, by William C. Conover, president of the Fuel Economizer Company. While the general theme of the lecture was the subject of combustion, it nevertheless had special application to the Cornell economizer, which was described and illustrated in our issue of April 5. We stated at that time that it was manufactured by the Cornell Economizer Company, but neglected to add that it was being marketed by the Fuel Economizer Company, 925 Chestnut street, Philadelphia.

Boiler Tube Failures

One of the tubes of a boiler in the power house of the Brooklyn Rapid Transit Company, Kent avenue and Division street, Williamsburg, N. Y., failed Sunday afternoon, April 10, injuring four men. Detailed information of the accident is not available.

It is reported in a Cleveland paper of recent date that the explosion of a boiler tube in the power house of the Oil Well Supply Company's Twenty-first street plant seriously injured three men, all of whom were badly scalded about the face and hands.

On April 10 the torpedo boat "Flusser" of the seventh torpedo flotilla, while making its final speed test, broke all records ever attained by naval vessels of the United States. A speed in excess of 35 miles an hour was maintained for four consecutive hours.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c each Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

TURBINE. Elmer E. Hauer, Springfield, Ohio, assignor to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 951,115.

COMBUSTION ENGINE. Charles E. Henriod, Neuilly-sur-Seine, France. 951,117.

TURBINE. Henry F. Weinland, Springfield, Ohio, assignor to the Lagonda Manufacturing Company, Springfield, Ohio, a Corporation of Ohio. 951,164.

MOTIVE-POWER APPARATUS. Charles H. Atkins, Springfield, Mass. 951,171.

INTERNAL-COMBUSTION ENGINE. Anthony Duffy, Philadelphia, Penn. 951,181.

EXPLOSIVE ENGINE. George J. Weber, Kansas City, Mo. 951,352.

GAS ENGINE. Giles W. Weller, Kimberly, Nev. 951,353.

TURBINE. Thomas S. Kemble, Cleveland, Ohio. 951,373.

ROTARY EXPLOSION ENGINE. Enrique J. Conill, Paris, France. 951,388.

ROTARY ENGINE. James M. Lively, New York, N. Y., assignor to William De L. Walbridge, New York, N. Y. 951,479.

CURRENT MOTOR. Felix H. Crago, Townsend, Mont. 951,525.

ROTARY STEAM ENGINE. Wilson R. Harper, Vinton, Iowa. 951,607.

CURRENT WATER WHEEL. Laban Lemert, Brownstown, Ill. 951,616.

ROTARY ENGINE. Justus R. Kinney, Dorchester, Mass. 951,735.

EXPLOSIVE ENGINE. George W. Smith, Henry, Mo. 951,747.

ROTARY ENGINE. Dawson Conekin, Tampa, Fla. 951,776.

BOILERS, FURNACES AND GAS PRODUCERS

OIL BURNER. Arthur E. Hauck, New York, N. Y. 951,394.

STEAM GENERATOR. James M. Lively, New York, N. Y., assignor to William De L. Walbridge, New York, N. Y. 951,480.

OIL BURNER. William R. Jeavons, Cleveland, Ohio. 951,612.

MECHANICAL STOKER. Charles W. Willey, Birnie, Manitoba, Canada. 951,712.

CRUDE-OIL BURNER. John A. Brabham, Frederick, Okla., assignor of two-thirds to Willis S. Robertson, Frederick, Okla., and Jesse B. Green, Charlestown, Tenn. 951,722.

POWER PLANT AUXILIARIES AND APPLIANCES

CONTINUOUS CARD INDICATOR. Amos P. Kinney, Quincy, Mass., assignor to American Steam Gauge and Valve Manufacturing Company, Boston, Mass., a Corporation of New Jersey. 951,131.

CENTRIFUGAL APPARATUS FOR COMPRESSING AND PROPELLING FLUIDS. Carlo Wedekind, St. Jean-sur-Mer, France. 951,162.

VALVE-CONTROLLING MECHANISM. Philip L. Wormeley, Washington, D. C. 951,168.

PACKING. Harry Brousseau, New York, N. Y., assignor of one-half to Marcus Stern, New York, N. Y. 951,175.

LUBRICATOR. Francis M. Cannon, McKeesport, Penn. 951,234.

GOVERNING MECHANISM FOR ELASTIC FLUID TURBINES. John G. Callan, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 951,301.

PUMP GOVERNOR. Lawrence Kjerulff, Kansas City, Mo. 951,318.

PACKING FOR PISTONS AND THE LIKE. George A. Pearse, Jr., New York, N. Y. 951,333.

GOVERNING MECHANISM FOR ELASTIC FLUID TURBINES. Richard H. Rice, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 951,337.

VALVE-CONTROLLING MECHANISM. John M. Roe, Chicago, Ill., assignor to Under-Feed Stoker Company of America, Chicago, Ill., a Corporation of New Jersey. 951,340.

PIPE COVERING. Walter Sherwood, North Yakima, Wash., assignor of one-half to John Sawbridge, North Yakima, Wash. 951,342.

NOZZLE FOR ELASTIC-FLUID TURBINES. George E. Stevens, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 951,346.

SHAFT PACKING. Walter Kieser, Berlin, Germany, assignor to General Electric Company, a Corporation of New York. 951,374.

VALVE. Paul A. Gstalder, Pittsburg, Penn., assignor of one-half to Charles P. Burke, Pittsburg, Penn. 951,392.

VALVE. George B. Moore, Springfield, Mass. 951,407.

PACKING RING FOR SHAFTINGS. Chas. G. Cunningham, Kings Bridge, N. Y. 951,435.

HOSE COUPLING. John H. Stephens, Vernon, Tex. 951,516.

VALVE GEAR. Harry O. Lawrence, St. Louis, Mo., assignor of two-thirds to John S. P. Gordon, East St. Louis, Ill., and one-third to Thomas P. Carroll, St. Louis, Mo. 951,536.

BALANCED VALVE. Collatinus A. Turner, New York, N. Y. 951,551.

CARBURETER. Gilman W. Brown, West Newbury, Mass. 951,590.

FRICTION CLUTCH. Van Zandt M. Moore, Cleveland, Ohio. 951,651.

PUMP. James M. Stratton, Salem, Ohio, assignor to the Deming Company, Salem, Ohio, a Corporation of Ohio. 951,663.

FEEDING MECHANISM FOR GAS PRODUCERS. John C. Swindell and John A. Swindell, Pittsburg, Penn. 951,665.

CALORIMETER. Arthur H. Walrath, Davenport, Iowa. 951,709.

ROTARY PUMP. Justus R. Kinney, Dorchester, Mass. 951,762.

ELECTRICAL APPARATUS AND APPLICATIONS

ELECTRIC LIGHTER. William Voltz, Chicago, Ill. 951,159.

BATTERY WELL. Charles F. Massey, Chicago, Ill. 951,192.

ELECTROLYTIC CELL. Jasper Whiting, Boston, Mass., assignor to the Whiting Company, Boston, Mass., a Corporation of Maine. 951,229.

ELECTRICAL CIRCUIT PROTECTOR. Charles A. Rolff, Adrian, Mich., assignor, by mesne assignments to Rolfe Electric Company, Rochester, N. Y., a Corporation of New York. 851,259.

State Conventions

POWER has written to the secretaries of the several State associations of the National Association of Stationary Enneers requesting particulars regarding their conventions for 1910. Up to the time of going to press the associations of Indiana, Texas and West Virginia have not replied. Following are the dates and places of the annual meetings of the State associations from whom we have heard:

Iowa, at Waterloo, May 5, 6 and 7.

Illinois, at Moline, May 21.

California, at Los Angeles, May 23 to 28.

Pennsylvania, at Scranton, June 3 and 4.

New Jersey, at Jersey City, June 4.

Colorado, at Denver, June 10.

Kentucky, at Hopkinsville, June 10 and 11.

New York, at Buffalo, June 10 and 11.

Ohio, at Hamilton, June 16, 17 and 18.

Wisconsin, at Oshkosh, June 17, 18 and 19.

Connecticut, at New Haven, June 24 and 25.

Massachusetts, at Lowell, July 7, 8 and 9.

Michigan, at Kalamazoo, July 21, 22 and 23.

Canadian Association of Stationary Engineers at Berlin, Ont., July 26, 27 and 28.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City. Spring meeting at Atlantic City, May 31 to June 6.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff, Denver, Colo.; sec. and treas., Frank M. Tait. Association York. Next annual convention, St. Louis, Mo., May 23-28.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor, 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, 803 Fulton building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., William G. Snow; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next convention, Philadelphia, Pa., June 6-10, 1910.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio. Next meeting, Cincinnati, May 19 and 20, 1910.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. E. Brown; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Niagara Falls, Canada, May 24-27, 1910.

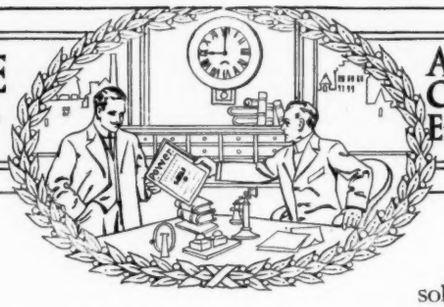
INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., A. C. Rogers, Toledo, O.; sec. and treas., D. L. Gaskill, Greenville, O. Next annual meeting at Toledo, O., May, 1910.

MOMENTS WITH THE Advertising Editor



A Department *for* Subscribers Conducted *by the* Business End *of* Power *and the* Engineer

Hugh Chalmers, of the Chalmers Motor Company, is one of the big *salesmen* of the country.

Therefore, he is a great believer in advertising.

Salesmen who are *big* in every sense know that selling and advertising are twin giants.

In the advertising bulletin of a recent number of Collier's Weekly, Chalmers talked of "The Responsibility of the Advertiser," and what he said will interest you because it will give you a clear idea of the advertisers' viewpoint.

"The first duty of a merchant or a manufacturer to the public is to advertise. It is his duty to advertise as well and as hard as he can, and to keep at it.

"In a broad, economic sense it is the duty of every man with something that can be advertised to advertise it as thoroughly as his means and knowledge will permit.

"The great problem with all of us is the problem of distribution. Every influence that aids distribution is a civilizing influence, for civilization is a matter of having the greatest possible amount of necessities, and luxuries, too, distributed among the greatest number of people. Advertising is the greatest single distributive force we know. The man who advertises and sells a good product confers a favor on the buyers by the very act of advertising and selling. If there is a new razor better than the one I have, I would like to know it. If a man can build a better typewriter or make a warmer coat, I would like to have him tell me. He is somehow short of his duty if he fails to tell me. Remember the Parable of the Talents.

"Many an article containing great possibilities for usefulness to the people has been largely lost to them because it was not properly pushed. It was not advertised.

"Advertising is the connecting link between invention and use by the public. Many a valuable invention has been lost to the world because the inventor either could not or did not tell the people what he had—because he did not convince them that they ought to buy it.

"If a product is a good product it deserves to be advertised. It is to the interest of *everyone* to have it advertised.

"The second duty of an advertiser to the public is to tell the absolute truth about the thing he has to sell. Untruthful advertising might succeed for a while, but it would fail in the end. There is no surer law than this. The only business that succeeds and keeps on succeeding nowadays is the business that is built on the groundwork of honest statement.

"The third duty of the advertiser to the public is to back up every honest claim made. It is the duty of the advertiser to take care of the people who buy his goods. It is his duty, especially if he is selling a specialty article, to help them in every way he possibly can to get the maximum amount of service out of the thing they have bought. The duty of the advertiser does not end with the sale. The only right kind of a sale is the one where the buyer and the seller mutually profit, and it is the seller's duty to help the buyer in every way he can to get all out of his purchase that he thought he was getting when he bought."

Much of what Chalmers says we have tried to express in these talks to our readers—only Chalmers says it better and more clearly.

One reason for the prosperity and activity of this country is because the people are good *spenders*.

We're not talking of the Willie Boys who dally with the lobster pots of Broadway.

We mean that all classes of business men and engineers are counted business men these days—are open minded to anything that will help them to get better results in their work.

They do not close their minds or pocket-books to any device that will prove profitable.

They take the broad view that it is the ultimate result, not the first cost, that counts.

Therefore, they are willing to be shown.

Ten years ago folks wrote in and told us there was too much advertising in POWER—they were paying for reading matter.

If we could and did cut out the advertising pages today, we'd be flooded with letters of protest—

From readers who know that it *pays* to read the advertising pages and buy advertised goods.